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INTERNAL COMBUSTION ENGINES

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INTERNAL COMBUSTION ENGINES

THEORY AND DESIGN

A TEXT BOOK ON GAS AND OIL ENGINES FOR ENGINEERS AND STUDENTS IN ENGINEERING

BY

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PREFACE TO SECOND EDITION

Any textbook dealing with present day practice is reliable only so long as changes are made in it to keep pace with the advance in the art with which the book deals. This is particularly true of a book on internal combustion engine practice. The author has felt for some time that a revision in this book was necessary in order to make it conform more closely to the gas engine situation today, but on account of his other duties found it almost impossible to revise the book, and make the revision as thorough and complete as it should be. Fortunately, assistance was offered by various men familiar with different phases of the work, and to the men the author is deeply indebted.

Among those who are largely responsible for the work are E. E. Adams, Chief Engineer, The Smith Gas Engineering Company, Dayton, Ohio, who wrote the chapter on Gas Producers; Professor Thomas E. Butterfield, Lehigh University, who wrote the chapter on Diesel and Oil Engines; Professor George A. Goodenough, University of Illinois, who rendered considerable assistance with the Thermodynamics in the earlier chapters; and Lawrence H. Pomeroy, who wrote the matter on automobile engine design, and assisted with other parts of the work.

It will be noticed that in certain cases descriptions of engines that are no longer manufactured have been allowed to remain in the book. This was done because it was thought that in each case the material illustrated good engineering practice and for that reason is valuable for instruction purposes.

R. L. S.

Pittsburgh, Pa.
August, 1923.

PREFACE TO FIRST EDITION

A LARGE part of the material in this book, especially in the portion dealing with design, has been used for several years by the author in the form of notes for a course in gas- and oil-engine design. These notes were added to from time to time until they became too bulky for use in typewritten form. The first thought of the author was to have the notes printed as such for private use only but it was finally decided to publish them in the form of a text book. In order to do this it was necessary to expand them and in doing this certain investigations and writings of other authors were consulted and drawn on for material without which a book of this nature would be valueless.

The book is written with the assumption on the part of the author that the reader has studied thermodynamics and mechanics. However, the author thought best to include in the book a review of the parts of thermodynamics which are absolutely essential to a clear understanding of the theory of internal combustion engines. Also, at certain places in the book, the author has digressed far enough to derive and explain certain indispensable relations in mechanics, not that they present anything new but because of a possibility that the reader has allowed them to become hazy from lack of use.

The author wishes to express his thanks to those manufacturers who kindly furnished drawings, photographs and data and to those publishers and authors who permitted their work to be used in the text of the book. He also expresses his thanks to his associates who assisted him in the preparation of drawings and particularly to Professor A. M. Greene, Jr., for his valuable aid and advice on portions of the book.

R. L. S.

RENSSELAER POLYTECHNIC INSTITUTE,
TROY, N.Y., *June* 17, 1915.

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INTERNAL COMBUSTION ENGINES

CHAPTER I

GENERAL

In writing this book the author assumes that it is to be used primarily as a text book, and that the student using it has a knowledge of mechanics and thermodynamics. Many students, however, who have a working knowledge of the subjects mentioned above are not at all familiar with the mechanical details of gas and steam engines. This chapter is devoted to giving the student a general idea of the details of gas engines, the names and functions of the various parts and the differences that exist between the gas engine and the steam engine.

One other assumption made by the author might well be explained at this point. No reference is made to the preparation of any kind of solid fuels in order to use them in gas-engines. In speaking of fuels for gas engines, then, it is with the assumption that the fuel is in the gaseous form, ready to be introduced into the engine.

Conversion of Energy. — The conversion of **chemical energy** into **mechanical energy** is accomplished by means of an engine, steam or gas. If a steam engine is used, an intermediate member is placed in the chain, this member being the boiler. The **chemical energy** of the fuel is changed into **heat energy** in the furnace and this heat energy is transferred to the water in the boiler. The **water** in this case acts as a carrier and does not enter vitally into the whole problem of heat conversion by means of the steam engine. That is to say, other carriers could be used in place of water, but it happens that water is plentiful and has certain characteristics which make its use as such a carrier desirable in many ways.

In the **engine cylinder** the heat in the steam is converted into **mechanical energy,** by moving a piston to and fro in this cyl-

inder. In some cases the reciprocating motion is used as such; in the great majority of engines, however, the reciprocating motion is changed into a rotary motion by means of a system of members forming a kinematic chain.

In the internal combustion engine **combustion** takes place directly in the **cylinder,** that is, the chemical energy is changed into heat energy in the cylinder, which acts, in this case, as the furnace of the boiler.

The **medium** or **carrier** in the internal combustion engine is formed from the **fuel** and the **air** required for combustion. This acts very nearly as a **perfect gas,** while the action of the steam in a steam engine cylinder is far from that of a perfect gas. The mechanical action of the steam engine is very similar to the internal combustion engine.

Advantages of a Gas Engine. — If the actions of a steam engine and an internal combustion engine are almost identical, why use one in preference to the other? The answer, theoretically, is this: — The **efficiency** of the Carnot cycle, the highest attainable, is $\frac{T_1 - T_2}{T_1}$, where T_1 is the absolute temperature of the high point of the cycle and T_2 is the temperature of the low point, or, in other words, the cycle works between limits of T_1 and T_2. The value of T_2 is fixed by the atmospheric temperature and is the same for both the internal combustion engine and the steam engine, assuming both these engines to operate on the Carnot cycle. In the steam engine, the value of T_1 is about 862 degrees for 250 pounds steam pressure. With a value of T_2 at 70° F. or 530 degrees absolute the efficiency would be 0.385. With a temperature of 2500° F. in the cylinder of the gas engine the efficiency would be 0.83 or more than twice that of a steam engine. Possibly, then, the gas engine has the advantage of the steam engine in the ratio of 0.83 to 0.385; but theoretical considerations alter this considerably. These considerations will be treated later in this book.

Heat Losses. — In the **steam-engine cycle** if the steam is cooled below the boiling point corresponding to its pressure some of it is condensed and the heat radiated is a direct loss. The problem, then, is to retain all the heat of the steam until the steam leaves the cylinder, or at least to prevent any radiation, so that the maximum amount of heat will be available for work.

Another, and greater loss, is the **heat** of **vaporization** at exhaust pressure. A large part of the heat that is transferred from the boiler furnace into the water in the boiler goes into the latent heat, or heat of vaporization. This heat is not available for doing work but is lost by being discharged into the atmosphere.

With the gas engine the reverse is true and, instead of retaining all the heat in the cylinder, some of it must be removed by conduction because temperatures are high enough to be injurious to the metal in contact with the hot gases. This excess heat is removed by means of jacket water.

The effect of **heat** on the gases in a gas engine cylinder is to increase the **pressure.** In order to reduce T_2 as low as possible and thus increase the efficiency the gases are expanded in the cylinder. To bring the gases in the cylinder down to atmospheric temperature **complete expansion,** or expansion down to zero pressure, would be necessary. Practically, that is not possible and the gases are exhausted at a pressure and temperature considerably above the temperature of the atmosphere. This is another large loss of heat which helps to decrease the theoretical efficiency.

Source of Power. — Inflammable gas or vapor is the source of heat in the gas engine cycle. The gas is **ignited** in the cylinder, **temperature increases** and the resulting **pressure** drives the **piston** forward. Many different kinds of gas, varying in heating value, are employed, and the effects obtained by ignition and explosion cannot be determined or understood without a knowledge of the chemical constituents of the gas, and the proportions in which they combine with the oxygen of the air. Since the gas does not contain the oxygen necessary for combustion, it can never be burned by itself, but must always be diluted with a certain quantity of air. Unless the composition of the gas and the ratio of its dilution with air are known, it is impossible to ascertain the temperatures and pressures in the cylinder and to calculate the work the gas ought to do. The value in design of the laws and phenomena of the combustion of gas and air mixtures will be taken up later in the part of this book devoted to the design.

Utilization of Power. — A brief study of the **construction** and **parts** of a gas engine is necessary in order to become familiar with the engine as a whole and the terms used in connection with

the various parts. We will enumerate these parts and then describe the functions they have to perform, together with the different operations taking place in a gas engine.

Base. — The **base-plate,** on which the engine is fixed and the cylinder bolted, is of **cast-iron,** usually, and of massive construction. This base-plate appears in a variety of forms, varying with the type of engine, use to which the engine is to be put and the manufacturer.

Cylinder. — The gas engine **cylinder,** made usually of **cast-iron,** is either horizontal or vertical, depending on the type of engine. In small engines only one single-acting cylinder is used, one end being open to the atmosphere. In larger engines two or more single- or double-acting cylinders are used. As one object to be attained in a gas engine is to have the gases expand to a low pressure, it appears that if the engine is made compound this object will be attained. This arrangement, though often tried, has never been found to be successful.

Gas can be **ignited** in an engine cylinder at **atmospheric pressure,** but as higher pressures and greater efficiency result if the gas is **compressed** previous to ignition, this method is used. In almost every case the gas is compressed in the working cylinder, one stroke of the cycle being devoted to this work.

A special feature of gas engine cylinders is that, on account of **high temperatures** reached, they are always provided with some means of **cooling** the walls. In the smallest sizes it has been found sufficient to increase the radiating surface by making the outer surfaces of the walls ribbed, exposing a large cooling area to the outside air. In larger engines this " air-cooling " is not sufficient, so the cylinders are cast with double walls, between which water is circulated to keep the inner wall cool. In single-acting engines the cooling effect is increased by the open end, the motion of the piston causing currents of air to rush into the cylinder at every revolution, thus greatly increasing the amount of heat conducted away from the piston.

Pistons. — The **pistons** of gas engines are very similar to steam-engine pistons except that they are made longer. In single-acting engines **trunk pistons** are used. Usually no special provision is made for cooling this type of piston as the circulation of air within the piston is very efficient. In **double-acting** engines, however, the pistons are cast hollow and are water-

cooled. All pistons are provided with Ramsbottom or snap rings to prevent leakage between the piston and cylinder walls.

Valves. — The **valves** of a gas engine perform the same operations that the valves of a steam engine do. They admit fresh gas into the cylinder at the proper time and allow the exhaust gases to escape. In addition, in some types of engines they assist in mixing the gases and in timing the explosions. In some of the earlier engines **slide-valves** were used to perform these functions, but as temperatures and pressures increased, the limitations of this type of valve were soon reached and poppet or lift valves were substituted. Occasionally, the fresh gas is admitted to the cylinder through cylindrical or piston valves but their use is not general.

In most gas-engines valves are worked from a **side-shaft** by **cams** or **eccentrics**; in others they are automatically lifted or closed by the pressure in the cylinder.

Transmission of Energy. — As in a steam engine the force on the reciprocating gas-engine piston is transmitted to the revolving **crank-shaft** through a **connecting-rod** and **crank.** In single-acting engines there is no piston-rod but the connecting-rod is pinned directly to the trunk piston which also acts as a cross-head. In double-acting engines the arrangement is identical with that of a steam engine, there being a **piston-rod, crosshead** and **connecting-rod.**

In all gas engines five operations are required for a complete cycle:

I. Admission and mixture of the gas and air.
II. Ignition.
III. Explosion or combustion.
IV. Expansion.
V. Exhaust, or discharge of the burned or inert gases.

To these another has been added in modern engines, namely, compression.

This **cycle** of work corresponds to one explosion but not necessarily to one revolution. In many engines the number of explosions and revolutions are independent of each other. The different cycles will be analyzed in a following chapter.

Admission of Gas and Air to the Cylinder. — In modern engines **admission** is effected with ordinary **lift valves,** both gas and air passing through the same valve. Before entering the

cylinder the gas usually passes through a chamber where it is thoroughly mixed with the proper proportion of air, which is admitted to the mixing chamber through a separate inlet. The valve which admits gas into the mixing chamber is usually connected to the **governor,** which regulates the quantity of gas entering, and consequently the number or strength of the explosions.

Ignition. — In the older types of engines **ignition** of the charge was done with an **open flame** which was located in a cavity separated from the cylinder by a valve, usually a slide-valve. At the proper time for ignition this slide-valve opened and put the flame into communication with the compressed charge in the cylinder proper. The explosion that followed usually extinguished the flame. To remedy this trouble a second valve was used to put this flame chamber into communication with a permanent flame outside the cylinder after the slide-valve leading from the flame chamber into the cylinder had closed.

The **flame ignition** has been superseded by others which are more reliable, chief among these being the **hot-tube, electric spark** and **combustion due to high compression temperatures.** The **hot-tube** ignition is similar to the flame in that it is placed in a separate chamber communicating with the cylinder proper by a valve which opens at the proper time. The tube which is closed at its inner end is kept hot by an external flame. The **electric ignition** is by means of a current from **primary batteries** or from a **magneto** or **dynamo.** The two systems of electric ignition, the "**make and break**" and the "**jump spark,**" will be dealt with later. Ignition by **high-compression temperature** is used in high-pressure oil engines where air is compressed to such a high pressure that the resulting temperature is above the ignition point of the fuel oil. Then when the fuel is sprayed into the cylinder, spontaneous combustion results. Obviously this system of ignition cannot be used with gaseous fuels.

Explosion and Expansion. — **Explosion** of the charge takes place in the combustion space of the cylinder as ignition occurs. In one type of engine that is now obsolete, ignition, and consequently explosion, took place when the piston had completed about one-half its suction stroke. Ignition thus occurred at atmospheric pressure. In order to increase the efficiency, compression was introduced. The pressure to which compression is now carried determines the relative volume of the combustion

or clearance space. The **point** of **ignition** in modern engines is determined by practical consideration but is usually at or near the dead center, or, in other words, when the piston is at or near the inner end of its stroke.

Exhaust. — Various methods have been employed to get rid of the **burned gases** after explosion, but practice has shown that they should be expelled from the cylinder as quickly and as completely as possible. In single-acting engines the exhaust-valve is closed during the forward suction stroke of the piston. In some engines it is open during half the return exhaust stroke, in others during the whole of this stroke, and in still others it is open during the last part of the forward stroke and the first part of the return stroke. In this last case the new charge is forced into the cylinder under pressure, displacing the burned gases.

The **exhaust-valve** of the modern gas engine works under severe conditions, as it opens when the pressure and the temperature are high. On account of this the heat lost in the exhaust is considerable. This is one of the defects of the modern engine which engineers are anxious to remedy.

Compression. — The one **improvement** which has done more than any other to make the modern gas engine successful is higher **compression.** It is done as follows: A charge of air and gas is drawn or forced into the cylinder and on the **return stroke** this charge is compressed into the **combustion space.** The volume of the combustion space is determined by the intensity of the compression. The advantage of this process is that the particles of gas are pressed more firmly together so that when ignition takes place the resulting temperature and pressure is higher than if the gas had not been compressed.

The advantages of compression are:

I. The **smaller size** of **cylinder** required. Since the pressure resulting from combustion is higher with compression than without, it follows that the total force on the piston is greater, consequently the work done is greater.

II. **Combustion** is more certain and more rapid than without compression because the particles of gas are forced close together, and ignition proceeds rapidly as the heat is communicated more rapidly from one particle to another.

III. **Greater economy** of gas because the resulting temperature is higher. It was shown earlier in this chapter that the

efficiency increased as the maximum temperature reached increases.

IV. **Poorer gas** can be used than without compression because combustion takes place so rapidly that the poor grade of gas will explode, whereas with no compression combustion would be so slow as to prevent the use of that gas in an engine.

Oiling. — The question of **lubrication** is of very great importance in a gas engine. Not only do we have the usual crank-pin, wrist-pin and shaft bearings to lubricate, but the piston requires the utmost care in this respect. In the steam-engine cylinder lubrication is not particularly difficult on account of the surfaces being wetted and the low temperature. In the gas engine, however, the rubbing surfaces are dry and the temperature and pressure are both high. On account of the high temperature the lubricating oil for the gas engine must be carefully selected.

Starting. — A **steam engine** is started by turning the steam under pressure into the cylinder. A gas engine cannot be started in this way because there is no pressure to be turned into the cylinder unless compressed air be used. **Large engines** are started with **compressed air** in exactly the same way that steam is used in a steam engine. Small engines are set in motion by hand or by an electric motor, this operation continuing until explosions start in the cylinder.

Speed Regulation. — **Speed regulation** is accomplished in one or two of several ways in a gas engine:

I. The **amount** of **mixture** may be cut down or increased to suit the load. This is **throttling.**

II. The **ratio** of **gas** to **air** may be changed, thus changing the power of the engine to suit the load. This is **constant quantity** governing.

III. The **mixture** may be **cut off** entirely for several working strokes in succession, the number depending on the load, thus preventing explosions or impulses for these strokes. This is the **"hit-and-miss"** method and is used only on small engines. The cutting off of the supply of mixture may be accomplished in various ways which will not be discussed at present. No matter what system of governing is used, the change of centrifugal force of a revolving weight, or the change of inertia of a weight, due to the change in speed, is used to actuate the mechanism which makes the proper change in speed, back to normal.

CHAPTER II

LAWS OF GASES — THERMAL LINES — PROCESSES — ENTROPY

While this book is written with the general understanding on the part of the author that the student already has a knowledge of thermodynamics, it will not be out of place at this point to review some of the more important laws of gases that will be used later on in the development of this work.

Boyle's Law. — At constant temperature the volume of a given weight of gas varies inversely as the pressure.

Charles' Law. — With the volume constant the change of pressure of a gas is proportional to the change of temperature.

By a combination of these two experimental laws the characteristic equation $PV = MBT$ may be deduced. P is the pressure of the gas under consideration in pounds per square foot, V is the volume in cubic feet, M, the weight of the gas in pounds, T, the absolute temperature in degrees Fahrenheit and B is a constant which may be found as follows:

$$B = \frac{PV}{T}, \text{ when } M = 1.$$

Substituting the known values for air at certain conditions of temperature and pressure, we have

$$B = \frac{2116.32}{0.08071 \times 491.6} = 53.34,$$

2116.32 = pressure per square foot with barometer at 760 millimeters,

$$V = \frac{1}{\text{wt. of 1 cu. ft. air}} = \frac{1}{0.08071},$$

$T = 491.6°$ F.

The value of B for any other gas may be found by using the weight per cubic foot of that gas in place of 0.08071. Thus for hydrogen

$$B = \frac{2116.32}{0.00559 \times 491.6} = 767.37.$$

For a **mixture** of gases the value of B_{mix} may be found by multiplying the fractional weight of each constituent by the value of B for that gas and adding the products. Thus for a producer gas with the following composition by weight:

$$\begin{aligned} H_2 &= 0.60 \text{ per cent} \\ CO &= 23.08 \\ CH_4 &= 1.44 \\ CO_2 &= 10.88 \\ N_2 &= 64.00 \\ &\overline{100.00} \text{ per cent} \end{aligned}$$

we have

$$\begin{aligned} 0.006 \times 767.37 &= 4.604 \\ 0.2308 \times 55.142 &= 12.727 \\ 0.0144 \times 96.314 &= 1.387 \\ 0.1088 \times 35.09 &= 3.820 \\ 0.64 \times 54.985 &= 35.190 \\ &\overline{57.748} \\ B_{mix} &= 57.748. \end{aligned}$$

The application of the general equation $PV = MBT$ may best be shown by the use of examples.

What will be the size of the gas holder required to contain 1200 pounds of the producer gas mentioned above, when the temperature is 82 degrees, gauge pressure 5 ounces per square inch?

$$V = \frac{MBT}{P},$$

$$P = 2116.32 + \tfrac{5}{16} \times 144 = 2161.32,$$
$$M = 1200,$$
$$B = 57.748,$$
$$T = 459.6 + 82 = 541.6,$$

$$V = \frac{1200 \times 57.748 \times 541.6}{2161.32} = 17{,}364 \text{ cubic feet.}$$

Characteristic Surface.— The characteristic equation $f(P, V, T) = 0$, having three variables, may be represented by a surface. A state or condition of a substance is defined by its coördinates P_1, V_1, T_1, and this condition is therefore represented by a point on the surface. If the condition changes, a second point with coördinates P_2, V_2, T_2 will represent the new condition. The succession of points between the initial and final conditions will be

represented by a succession of points on the surface. The surface representing the equation $PV = BT$ is shown in Fig. 1.

Thermal Lines. — If, during a change of condition, the **temperature** of a substance shall remain **constant,** the point will evidently move on the characteristic surface parallel to the PV-plane. Such a change of condition is called an **isothermal,** and the curve described by the point is the isothermal curve. By taking different constant values for the temperature, we get a complete representation of the characteristic equation. For the perfect gas the isothermal lines consist of a system of **equilateral hyperbolas** having the general equation

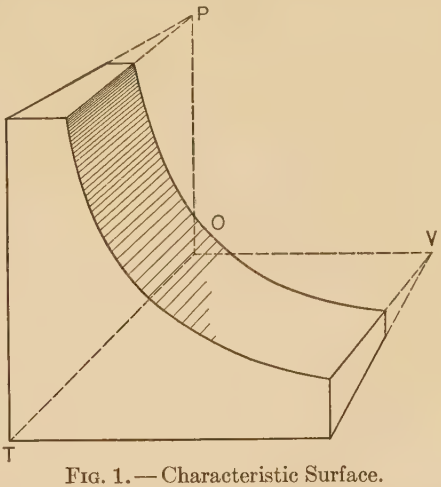


FIG. 1. — Characteristic Surface.

$$PV = \text{constant}.$$

If the **pressure** remains constant during a change of condition, the point will move parallel to the TV-plane and the projection of the path on the PV-plane will be a *straight* line parallel to OV, as AB in Fig. 2.

If the substance changes its condition at **constant volume,** the point moves parallel to the PT-plane and the projection of the path on the PV-plane is a **straight line** parallel to the P-axis as CD, Fig. 2.

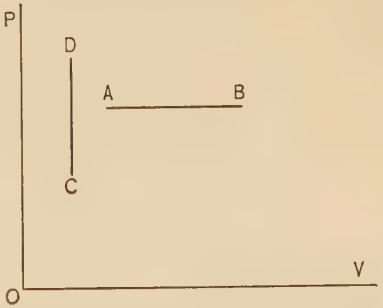


FIG. 2. — Constant Pressure and Volume Lines on PV-Plane.

Besides the cases just given, there are others which are not of prime importance in gas-engine work. One, however, is of great importance and will be taken up later on. This is the **adiabatic,** representing a change of condition in which the system neither receives nor gives out heat.

Specific Heat. — In general, the heat required to raise the temperature of a body 1 degree under given external conditions is called the **thermal capacity** of the body. Thus the thermal capacity of 1 gram of water at 17.5° C. is 1 calorie, and that of 1 pound of water at 63.5° F. is 1 British thermal unit.

The **specific heat** of a substance at a given temperature is the ratio of the thermal capacity of that substance at this temperature to the thermal capacity of an equal mass of water at some chosen standard temperature. If we take 17.5° C. (63.5° F.) as the standard temperature, and denote by L the thermal capacity per unit weight, then the specific heat c is given by the relation

$$c = \frac{L_t \text{ (of substance)}}{L_{17.5} \text{ (of water)}}.$$

For water $L_{17.5} = 1$ calorie. It follows, then, that the specific heat at the temperature t is numerically equal to the thermal capacity of a unit weight of the substance at the same temperature; thus, at 100° C. the thermal capacity of 1 gram of water is found to be 1.005 calories, and the specific heat is

$$\frac{L_{100}}{L_{17.5}} = \frac{1.005 \text{ cal.}}{1 \text{ cal.}} = 1.005.$$

Adiabatic Processes. — When a system, in changing its condition, has no thermal communication with other bodies, and, therefore, neither absorbs nor gives out heat, the change of condition is said to be **adiabatic.** In general, adiabatic changes are possible only when the system is enclosed in a non-conducting envelope. Rapid changes of condition are approximately adiabatic since time is required for conduction or radiation of heat; thus, the compression of air in an " air-cooled " compressor cylinder is practically adiabatic as the time is so short that little heat enters the cylinder walls. Since there is no heat added during adiabatic compression nor taken away during adiabatic expansion, the **work** done on the system during compression or by the system during expansion is at the expense of the **intrinsic energy.**

The projection on the PV-plane of the path of a point during an adiabatic change of condition gives the **adiabatic** curve. This curve is somewhat steeper than the isothermal curve as is shown by Fig. 3, where AB is the adiabatic and CD the isothermal. This follows from the fact that during adiabatic expansion the energy

decreases and as a result the temperature falls; hence for the same final volume the temperature, and therefore the pressure, is lower for the adiabatic expansion than for the isothermal expansion.

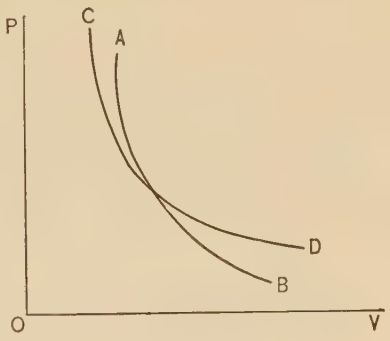


FIG. 3.—Adiabatic and Isothermal Curves on PV-Plane.

Availability of Energy. — Either **mechanical** or **electrical** energy can be completely transformed into the other in theory and nearly so in practice. Either one can be completely transformed into **heat.** Examples of these changes are the dynamo driven by a steam or gas engine for the change from mechanical to electrical, and the prony brake used for absorbing the power of an engine or the rheostat used for absorbing the power of a dynamo for the change from mechanical and electrical to heat energy. On the other hand experience shows that heat energy is not capable of complete conversion into mechanical work, and to get even a part of heat energy transformed into mechanical energy, certain conditions must be satisfied. As a first condition there must be two bodies of different temperatures; it is impossible to derive work from the heat of a body unless there is available a second body of lower temperature. Even if we have two bodies of different temperatures and place them in contact, the heat will flow from the hotter to the colder and no work will be obtained. Hence, as a second condition, the systems or bodies of different temperatures must be kept apart and a third system used to convey the energy. This third system may be called the medium. In the steam plant, for instance, the boiler furnace is the system of high temperature, the condenser is the system of low temperature and the steam is the medium. This medium is placed in contact with the hot system and receives heat from it; then by a change in condition expansion gives up energy in the form of work and delivers to the system of low temperature an amount of heat smaller than the amount taken from the source of high temperature, the difference being transformed into work.

If the total heat taken from the furnace be called Q_1 and the part

rejected into the condenser Q_2, then the heat transformed into work will be $Q_1 - Q_2$. This difference is also the available part of Q_1, and the part that is rejected into the condenser is the unavailable part or the waste. The ratio $\frac{Q_1 - Q_2}{Q_1}$ is called the availability of Q_1 for the transformation into mechanical energy. In general the term signifies the fraction of the energy of a given system in a given state or condition that can be transformed into mechanical work.

Reversibility. — A process is said to be reversible when the following conditions are fulfilled:

1. When the direction of the process is reversed, the system taking part in the process can assume in inverse order the conditions traversed in the direct process.

2. The external actions are the same for the direct and reversed processes or differ by an infinitesimal amount only.

3. Not only the system undergoing the change but all the connected systems can be restored to initial conditions.

A process that fails to meet these requirements in any particular is an irreversible process.

Suppose a cylinder confines gas to act on a piston, as in a steam or gas engine. See Fig. 4. If A is the piston area, the pressure acting on the face of the piston is pA, and for equilibrium this pressure must be equal to the force F. If we now assume that the force pA is slightly greater than F, the piston will move slowly to the right and the confined gas will assume a succession of conditions indicated by the curve AB. If at the condition B the motion is arrested and F is made infinitesimally greater than pA for all positions of the piston, the series of conditions from B to A will be retraced and the system (the confined gas in this case) will be brought back to its original state without leaving changes in outside bodies. The reverse process is accomplished by an infinitely small modification of the external force F. Therefore the process is reversible.

As examples of **irreversible** processes we might look at the following:

1. **Friction,** as in a **journal.** When a journal turns in a bearing, friction is always present and the work of friction is transformed into heat which is radiated. It would be folly, however, to try to turn the journal by heating the bearing. Thus the process is irreversible.

2. **Conduction** in **partitions.** We have all experienced the flow of heat from a warm room into the colder air but no one has yet been fortunate enough to experience the flow of heat from a cold atmosphere into a warm room. Hence the irreversibility.

Second Law of Thermodynamics. — According to the **First Law** of **Thermodynamics** the total quantity of energy in a system of bodies cannot be increased nor decreased by any change, reversible or irreversible, that may occur within the system. The

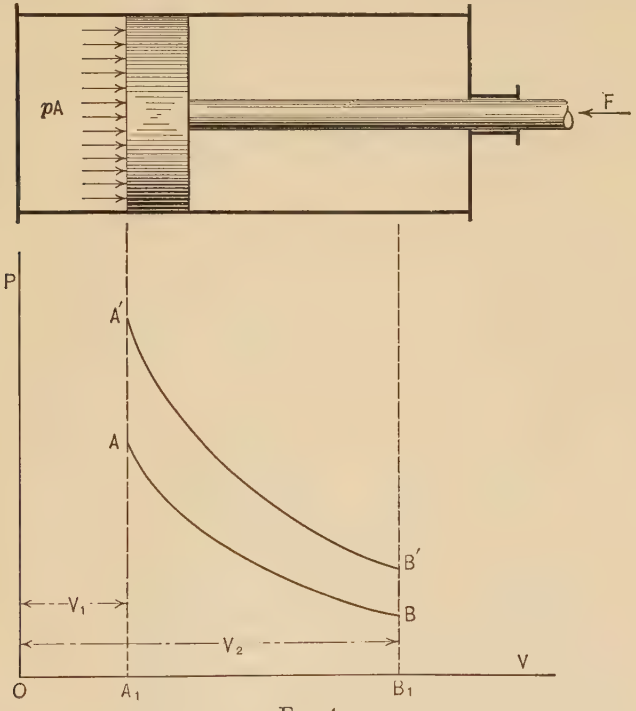


FIG. 4.

total energy, however, is not of as great importance as the available energy, and experience shows that a change within the system usually results in a change in the availability of the energy of the system.

Professor Goodenough, in his "Principles of Thermodynamics," deduces the following general laws to explain the **Second Law** of **Thermodynamics** and the **degradation** of **energy:**

1. No change in a system of bodies that can take place of itself can increase the available energy of the system.

2. An irreversible change causes a loss of availability.
3. A reversible change does not affect the availability.

Carnot's Cycle. — Keeping in mind the statements of the preceding article we may now look at the cycle of processes first described by Carnot in 1824. A cycle has been defined as a series of operations at the end of which the working substance reaches initial conditions. The Carnot cycle represents the best that can be done in the conversion of heat energy into mechanical energy. It cannot be used in an actual engine but is of theoretical interest as a criterion of the best results obtainable. The apparatus for this cycle consists of a source of heat S, at a temperature T_1 a refrigerator at a temperature T_2; and a working fluid or medium M.

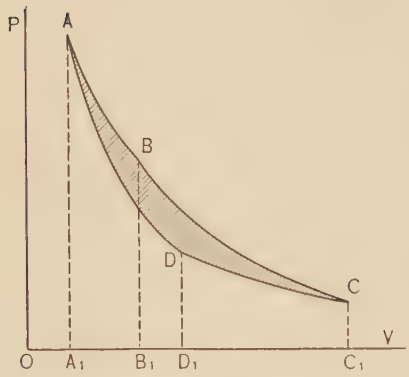


Fig. 5.

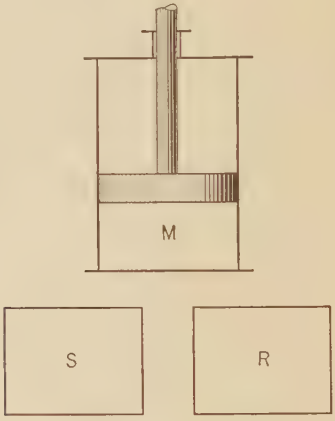


Fig. 6.

Assume the medium to be initially at a condition shown by B, Fig. 5, at the temperature T_1 of the reservoir S, then expand adiabatically until its temperature falls to T_2, the temperature of the body R. By this expansion the second condition C is reached and the work done by the medium is represented by BCC_1B_1. The expansion is assumed to proceed slowly so that the pressures on the two faces of the piston are equal, and the process is therefore reversible. The cylinder is now placed in contact with R so that heat can flow from M to R and the medium is compressed. The work represented by C_1CDD_1 is done on the medium, and the heat Q_2 passes from the medium to the refrigerator. The process is

again assumed to be so slow as to be reversible. From the condition D the medium is now **compressed adiabatically,** the cylinder being removed from R until its temperature again becomes T_1, that of the source S. During the third process work represented by D_1DAA_1 is done on the fluid. Finally the cylinder is placed in contact with S and the fluid allowed to expand at the constant temperature T_1 to the initial condition B. Work represented by the area A_1ABB_1 is done by the fluid during the process, and the temperature is kept constant by the flow of heat Q_1 from S to M.

The area $ABCD$ enclosed by the four curves of the cycle represents the **mechanical work** gained; that is, the excess of the work done by the medium over the work done on the medium. Denoting this by W we have $Q_1 - Q_2 = \frac{W}{J}$, where J is Joule's equivalent.

The **efficiency** of the cycle is $\frac{Q_1 - Q_2}{Q_1}$ and is equal to $\frac{T_1 - T_2}{T_1}$.

Since all processes of the Carnot cycle are reversible, it is evident that they may be traversed in reverse order. Carnot set forth the principle that of all engines working between the same source and the same refrigerator, no engine can have a greater efficiency than that of the reversible engine, and thus the efficiency is independent of the working fluid.

Available Energy and Waste. — Carnot's cycle gives us a means of measuring the **available energy** of a system and the waste due to an irreversible change of condition. Suppose a quantity of heat ΔQ is absorbed by the system at a temperature T, and we wish to find the part of this heat that can be transformed into work. No device can transform a larger portion of heat ΔQ into work than the Carnot engine. If T_0 is the lowest temperature that can be obtained for a refrigerator, the fraction $\frac{T - T_0}{T}$ of Q can be transformed into work by the Carnot engine and this may be called the available heat, or the available part of ΔQ is $\Delta Q \left(1 - \frac{T_0}{T}\right)$; the unavailable part is $\Delta Q - \Delta Q \left(1 - \frac{T_0}{T}\right) = \Delta Q \frac{T_0}{T}$.

For conduction of a quantity of heat Q from a temperature of T_1 to a temperature of T_2, it may be shown that the unavailable energy is increased by an amount $T_0 \left(\frac{Q}{T_2} - \frac{Q}{T_1}\right)$.

During an irreversible conversion of work into heat through the agency of friction, the amount of heat thus generated we may call ΔQ. $\frac{T - T_0}{T}\Delta Q = \left(1 - \frac{T_0}{T}\right)\Delta Q$ may be transformed back into work. Of the work $J\Delta Q$ expended in producing heat ΔQ, the part $J\Delta Q\left(1 - \frac{T_0}{T}\right)$ may be recovered in the form of work. The remainder, $JT_0\frac{\Delta Q}{T}$ is rendered unavailable.

Entropy. — All the expressions for the increase in unavailable energy derived above have a common factor T_0. From this it appears that the available energy varies with the temperature of the coldest refrigerator we may have and that the lower T_0, the less the unavailable heat.

The other factor has the form $\frac{Q}{T}$, or in some cases $\int\frac{dQ}{T}$. To this second factor, which, when multiplied by T_0 gives as a product the increase of unavailable energy, the name increase of entropy is given. The change of entropy of a system is therefore a measure of the change of unavailable energy of the system; an increase of entropy involves an increase of unavailable energy and vice versa. Entropy may then be defined as follows:

If, from any cause whatever, the unavailable energy of a system is increased and the increase is divided by T_0, the lowest temperature available for a cold body, the quotient is the increase of entropy of the system.

The "system" spoken of may be either a single substance, as the medium employed in a heat motor, or it may be all the bodies taking part in the process. When we apply the idea of increase of entropy to the system composed of all the bodies involved in the process, we are led to the conception that the increase of entropy measures the degradation of energy of the process. If this notion is combined with that expressed in the second law, we arrive at the following principles:

1. Any process that can proceed of itself is accompanied by an increase of the entropy of the system of bodies involved in the process.

2. The direction of a process, physical or chemical, that occurs of itself is such as will bring about an increase of entropy in the system.

Another definition of entropy is that derived by considering only one of the bodies taking part in a process, say the medium M in the Carnot cycle.

The change of entropy of a system corresponding to a change of the system from condition 1 to condition 2 is the definite integral $\int_{T_1}^{T_2} \frac{dQ}{T}$ taken along any reversible path between the two conditions.

The entropy thus defined is purely relative. We are never concerned with the absolute value of the entropy of a system in a given condition; what is desired is the change of entropy corresponding to a given change of condition. For the convenience of calculation we may assume the zero of entropy to be the entropy of a system in some specified state.

Temperature-entropy Plane. — Since the entropy of a system measured from an arbitrary zero is dependent only on the condition of the system, that is, the entropy is a function of the coördinates of the system, entropy may thus be included in the coördinates used to define a system and used with the other coördinates to form pairs. For convenience we use it with temperature only, giving us, in addition to the PV-plane, the TS-plane. The other planes formed by combinations of the other coördinates are not of especial importance in gas-engine design.

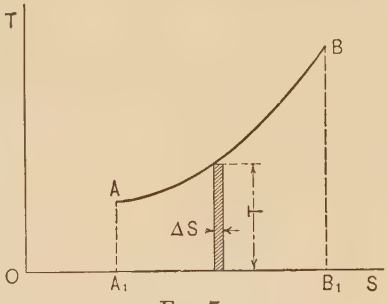


FIG. 7.

From the second definition of entropy we have

$$S_1 - S_2 = \int_{T_2}^{T_1} \frac{dQ}{T}, \tag{1}$$

or

$$dS = \frac{dQ}{T}, \tag{2}$$

and

$$T\,dS = dQ. \tag{3}$$

Let the curve AB, Fig. 7, be the path of a point projected on the TS-plane, the points A and B representing respectively the initial and final conditions. The area A_1ABB_1 represents the limit of the sums of the terms of the type $T\Delta S$, that is, the area

under the curve AB is the definite integral $\int_{S_1}^{S_2} T\,dS$. From equation (3) this is equal to the heat absorbed by the system from external sources during the change. One restriction must be made. *The change in condition must not involve any internal irreversible effects. In other words, the area under AB is equal to the heat absorbed only when AB is reversible.*

EXERCISES

1. What would be the weight of 1350 cu. ft. of air at 112° F., 102.3 lbs. gauge pressure?

2. Find the value of B_{mix} for a gas of the following composition by weight:

	Per cent
H_2 =	4.50
CO =	26.00
CH_4 =	2.43
CO_2 =	21.50
N_2 =	45 57
Air =	
	100 00

3. Find the height of a tank 25. ft. in diameter, to hold 1250 lbs. of the gas in Prob. 2. Pressure 8 oz. per sq. in., temperature 80° F.

CHAPTER III

PRINCIPAL GAS-ENGINE CYCLES

Carnot — Brayton — Lenoir — Diesel — Otto

General. — Many cycles have been proposed and experimented on for internal combustion engine work but few have proved to be practical. One of historical and theoretical interest is the **Carnot** cycle mentioned in the last chapter. This cycle has never been realized in actual practice, although many experimenters have attempted to build engines to use it. The **Lenoir** cycle was first successfully used about 1860 and many engines were built to use that cycle. It proved to be low in actual efficiency and was abandoned when the **Beau de Rochas** or **Otto** cycle came into use about fifteen years afterward. The **Otto** may be used on either the two-stroke or four-stroke principle. If the four-stroke principle is used, the exhaust and suction strokes do not enter into the thermal analysis of the cycle. When the two-stroke principle is used, these events take place outside the cylinder and consequently have no bearing on the thermal analysis of the cycle. The **Brayton** cycle was introduced in the United States in 1873. This cycle was composed of a **constant-pressure** line where **heat** was **added** by the burning of the mixture, **adiabatic expansion** to the **atmospheric pressure, exhaust** or **discharge** of heat at **constant pressure,** and **adiabatic compression.** This cycle was used for a short time and discarded. The **Diesel** cycle is somewhat similar to the **Brayton** in that the heat is added at **constant pressure** before **adiabatic expansion.** Heat is **discharged,** however, at **constant volume** instead of at **constant pressure** as in the **Brayton** cycle. **Diesel** first tried to use the **Carnot** cycle but found the necessary pressures and resulting temperatures too high for practical use.

Carnot Cycle. — Referring to Fig. 5, the heat **added** during **isothermal expansion** AB is

$$Q_{ab} = \frac{1}{J} P_a V_a \log_e \frac{V_b}{V_a}, \qquad (1)$$

and the heat given up or exhausted is

$$Q_{cd} = \frac{1}{J} P_c V_c \log_e \frac{V_d}{V_c} = -\frac{1}{J} P_c V_c \log_e \frac{V_c}{V_d}. \tag{2}$$

The heat transformed into work is then

$$\frac{W}{J} = Q_{ab} + Q_{cd} = \frac{1}{J}\left(P_a V_a \log_e \frac{V_b}{V_a} - P_c V_c \log_e \frac{V_c}{V_d}\right). \tag{3}$$

Since the temperature at A is T_1,

$$P_a V_a = MBT_1 \tag{4}$$

and

$$P_c V_c = MBT_2. \tag{5}$$

Since BC is an **adiabatic** we have

$$\left(\frac{V_c}{V_b}\right)^{n-1} = \frac{T_1}{T_2}, \tag{6}$$

and for the adiabatic DA,

$$\left(\frac{V_d}{V_a}\right)^{n-1} = \frac{T_1}{T_2}. \tag{7}$$

Then from (6) and (7)

$$\frac{V_c}{V_b} = \frac{V_d}{V_a} \quad \text{or} \quad \frac{V_c}{V_d} = \frac{V_b}{V_a}. \tag{8}$$

Inserting the values found in (4), (5) and (8) in (3) we get

$$\frac{W}{J} = \frac{MB}{J}(T_1 - T_2) \log_e \frac{V_b}{V_a}. \tag{9}$$

Then

$$E = \frac{\frac{W}{J}}{Q_{ab}} = \frac{MB(T_1 - T_2) \log_e \frac{V_b}{V_a}}{MBT_1 \log_e \frac{V_b}{V_a}} = \frac{T_1 - T_2}{T_1}. \tag{10}$$

It is important to notice here that T_1 is the **highest** and T_2 the **lowest** temperature reached in the cycle.

On the ***TS*-plane** the **Carnot** cycle will be in the form of a **rectangle,** as shown in Fig. 8. Heat is added at **constant temperature** along AB. **Adiabatic expansion** takes place from B to C, **heat** is **exhausted** along CD and **adiabatic compression** occurs from D to A.

Since the **processes** in this cycle are **reversible,** the area under the lines represents the heat added from an external source. The **heat added** along AB is then

$$(S_1 - S_2) T_1. \tag{1}$$

The heat rejected along CD is

$$(S_1 - S_2)\, T_2. \tag{2}$$

The **heat transformed** into work is the **difference**

$$(S_1 - S_2)\,(T_1 - T_2). \tag{3}$$

The **efficiency** is, by definition (3) divided by (1),

$$\frac{(S_1 - S_2)}{(S_1 - S_2)}\,\frac{(T_1 - T_2)}{T_1} = \frac{T_1 - T_2}{T_1}. \tag{4}$$

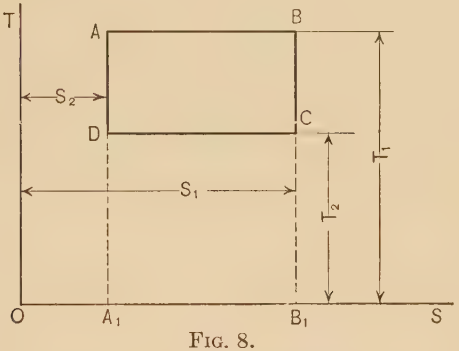


FIG. 8.

The Lenoir Cycle. — In the **Lenoir cycle combustion** is effected **without** previous compression. The charge is drawn into the cylinder during the **first** part of the stroke and exploded at a point suitable to the load. The pressure then **rises,** due to **heating** of the charge at approximately **constant volume.**

Figs. 9 and 10 represent, respectively, the PV- and TS-diagrams of this cycle.

From B to C the heat is **added** to the charge by **combustion** at **constant volume.** This heat is

$$Q_1 = c_v\,(T_c - T_b). \tag{1}$$

The heat rejected from A to B is

$$Q_2 = c_p\,(T_a - T_b). \tag{2}$$

The **efficiency,** being the heat transformed into work divided by the heat supplied, is

$$E = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{c_p\,(T_a - T_b)}{c_v\,(T_c - T_b)} = 1 - n\,\frac{\dfrac{T_a}{T_b} - 1}{\dfrac{T_c}{T_b} - 1} \tag{3}$$

since $\frac{c_p}{c_v} = n$. During the period of **constant pressure** from A to B

$$\frac{T_a}{T_b} = \frac{V_a}{V_b}, \tag{4}$$

and during the period of **constant volume** from B to C

$$\frac{T_c}{T_b} = \frac{P_c}{P_b}. \tag{5}$$

Also from the relations of **pressure** and **volume** along an **adiabatic**

$$\frac{P_c}{P_a} = \left(\frac{V_a}{V_c}\right)^n. \tag{6}$$

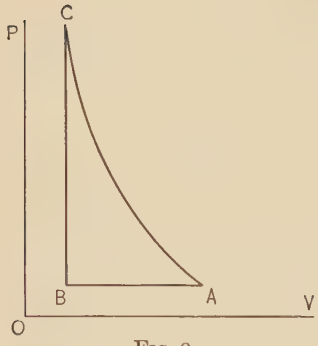


FIG. 9.

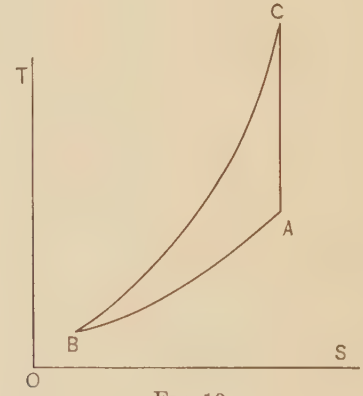


FIG. 10.

Lenoir Diagram.

But as $P_a = P_b$ and $V_c = V_b$,

$$\frac{P_c}{P_b} = \frac{T_c}{T_b} = \left(\frac{V_a}{V_b}\right)^n. \tag{7}$$

Let $\frac{V_a}{V_b} = r$, the **ratio** of **expansion;**

then

$$\frac{T_c}{T_b} = r^n, \tag{8}$$

and the efficiency is

$$E = 1 - n\frac{r-1}{r^n - 1}. \tag{9}$$

By transformation of equation (1) we get

$$\frac{T_c}{T_b} = \frac{Q_1}{c_v T_b} + 1 = r^n, \tag{10}$$

for the **range** of **temperature** during the cycle.

The **maximum pressure** reached will be

$$P_c = r^n P_b = P_b\left(\frac{Q_1}{c_v T_b} + 1\right). \qquad (11)$$

The Brayton Cycle. — In the **Brayton engine** the mixture of air and gas was compressed into a reservoir to a pressure of about 60 pounds per square inch and from this reservoir it flowed into the working cylinder, where it was **ignited** by a **flame.** A wire gauze **diaphragm** was used to prevent the **flame flashing back** into the reservoir. The mixture was thus burned in the working cylinder, **without explosion,** during about one-half the stroke of the piston. By proper regulation of the admission valve the rate of combustion was regulated to give practically **constant pressure** during **combustion.**

The ideal cycle of operations is as shown in Fig. 11.

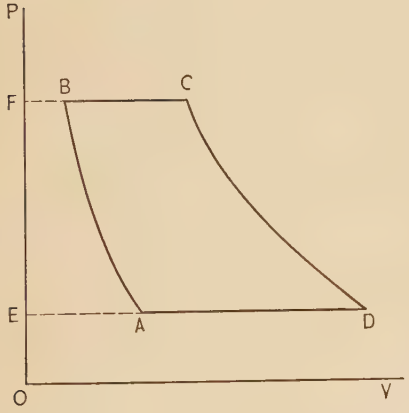


FIG. 11.

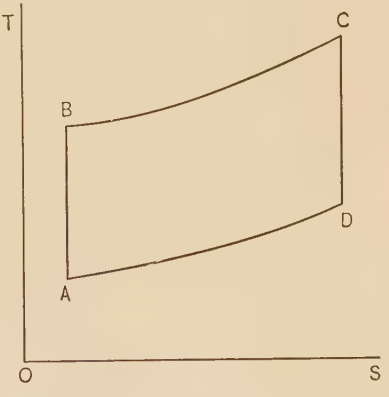


FIG. 12.

Brayton Diagram.

From E to A the charge is drawn into the **compression** cylinder.

From A to B **adiabatic compression** occurs.

From B to C **expulsion** from compressor occurs **simultaneously** with **admission** and **combustion** in motor cylinder.

From C to D **adiabatic expansion** takes place after **cut-off.**

From D to E **burned gases** are **expelled.**

The area $EABF$ represents the negative work of the **compressor** and the area $EDCF$ the **work obtained** from the **motor;** hence, the area $ADCB$ represents the **net available** work.

$$\text{Heat generated } Q_{bc} = Mc_p\,(T_c - T_b). \tag{1}$$

$$\text{Heat discharged } Q_{da} = Mc_p\,(T_d - T_a). \tag{2}$$

$$\text{Work done in form of heat energy} = Q_{bc} - Q_{da} = Mc_p\,(T_c - T_b - T_d + T_a). \tag{3}$$

$$E = \frac{Mc_p\,(T_c - T_b - T_d + T_a)}{Mc_p\,(T_c - T_b)} = 1 - \frac{T_d - T_a}{T_c - T_b} = 1 - \frac{T_a}{T_b}\cdot\frac{\frac{T_d}{T_a} - 1}{\frac{T_c}{T_b} - 1}. \tag{4}$$

Since the pressures are constant, we have

$$\frac{T_c}{T_b} = \frac{V_c}{V_b} \quad \text{and} \quad \frac{T_d}{T_a} = \frac{V_d}{V_a}.$$

Also from the adiabatic relation we have

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{n-1},$$

$$P_b = \left(\frac{V_a}{V_b}\right)^n P_a,$$

$$P_c = \left(\frac{V_d}{V_c}\right)^n P_d.$$

Since $P_b = P_c$ and $P_a = P_d$,

therefore $$\frac{V_a}{V_b} = \frac{V_d}{V_c},$$

and $$\frac{V_c}{V_b} = \frac{V_d}{V_a} = \frac{T_c}{T_b} = \frac{T_d}{T_a}.$$

Hence $$E = 1 - \frac{T_a}{T_b} = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} = 1 - \frac{1}{r^{n-1}}, \tag{5}$$

where $$r = \frac{V_a}{V_b}.$$

The **temperature range** may be found as follows:

$$\frac{T_c}{T_b} = \frac{Q_{bc}}{c_p T_b} + 1. \quad [\text{(from Eq. (1)}]$$

$$\frac{T_b}{T_a} = \left(\frac{V_a}{V_b}\right)^{n-1} = r^{n-1},$$

or $$T_b = T_a r^{n-1};$$

hence $$\frac{T_c}{T_a} = r^{n-1}\left(\frac{Q_{bc}}{c_p T_b} + 1\right). \tag{6}$$

The **maximum pressure** will be

$$P_c = P_a r^n. \tag{7}$$

The *TS*-diagram of the ideal **Brayton** cycle is as shown in Fig. 12. It will be noticed later that the appearance is very much the same as the *TS*-diagram of the **Otto** cycle, the difference being that the lines connecting the **adiabatics** in the **Brayton** cycle are **constant-pressure** lines while in the **Otto** they are **constant-volume** lines.

The Diesel Cycle. — In the **Diesel** engine **air** without fuel is **compressed** to a pressure of about 500 pounds per square inch. The **temperature** at the end of compression is consequently higher than the **ignition** temperature of the fuel. At the end of the compression stroke fuel is injected into the compressed air and burns immediately. By proper **regulation** of the **fuel valve**

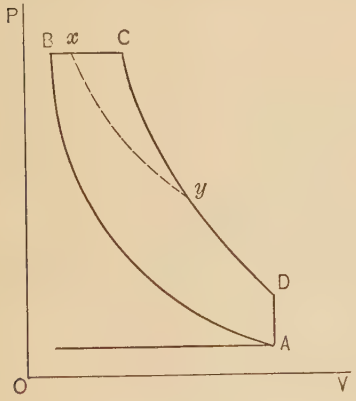


Fig. 13.

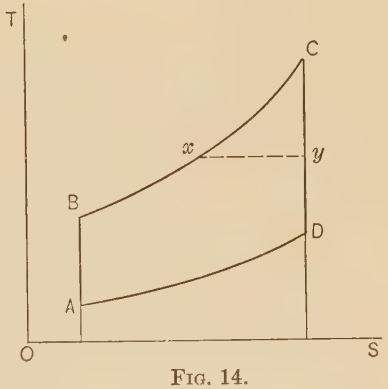


Fig. 14.

Diesel Diagram.

the air may be made to expand for a short period at **constant pressure,** or, if desired, at a **falling pressure** and approximately **constant temperature.** As in the **Brayton** engine **governing** is effected by cutting off the fuel supply earlier or later.

The ***PV*-diagram** of the ideal **Diesel** cycle is shown in Fig. 13. It was the **original** aim of **Diesel** to regulate the fuel supply so that a short period of **combustion** *Bx* would be followed by **isothermal expansion** *xy*. He found, however, that in order to get this result it would be necessary to compress the air to a **higher pressure** than could be taken care of in **practice.** The **mechanical difficulties** were so great that they could not be overcome in an engine that would be practical. The cycle that was finally brought out by the late Dr. Diesel is *ABCD*.

On the **TS-plane** this cycle appears as in Fig. 14. BC is a **constant-pressure** curve, CD **adiabatic expansion,** DA **constant volume** and AB **adiabatic compression.**

The **heat added** from B to C is

$$Q_{bc} = Mc_p\,(T_c - T_b). \tag{1}$$

The **heat rejected** from D to A is

$$Q_{da} = Mc_v\,(T_d - T_a). \tag{2}$$

Heat **transformed into work** is

$$Q_{bc} - Q_{da} = M\,[c_p\,(T_c - T_b) - c_v\,(T_d - T_a)]. \tag{3}$$

$$E = \frac{c_p\,(T_c - T_b) - c_v\,(T_d - T_a)}{c_p\,(T_c - T_b)} = 1 - \frac{1}{n}\frac{(T_d - T_a)}{(T_c - T_b)}, \tag{4}$$

where $n = \dfrac{c_p}{c_v}$.

The Otto Cycle.—This cycle, which is of utmost importance on account of its wide adoption for gas and oil engines, is shown on the PV-plane in Fig. 15.

From E to A the **explosive mixture** is drawn into the cylinder.

From A to B the mixture is **compressed adiabatically.**

From B to C the mixture is **ignited** and burned at **constant volume.**

From C to D the gases **expand adiabatically.**

From D to A heat is **discharged** at **constant volume.**

From A to E the inert gases are swept out of the cylinder at **constant pressure** by the piston.

This description fits the four-stroke cycle. In the two-stroke cycle the operations EA and AE are performed in a separate cylinder, but as they do not enter into the thermal analysis, such an analysis for the four-stroke would apply also to the two-stroke cycle.

This cycle appears on the TS-plane as shown in Fig. 16. BC and AD are **constant-volume** lines and AB and CD are **adiabatics.** Heat is added from B to C and if we assume c_v to be constant, we have

$$Q_{bc} = Mc_v\,(T_c - T_b) \tag{1}$$

and the **heat rejected** from D to A

$$Q_{da} = Mc_v\,(T_d - T_a). \tag{2}$$

Heat transformed into **work** is the difference of these two or

$$Q_{bc} - Q_{da} = Mc_v(T_c - T_b - T_d + T_a), \tag{3}$$

$$E = \frac{Mc_v(T_c - T_b - T_d + T_a)}{Mc_v(T_c - T_b)} = 1 - \frac{T_d - T_a}{T_c - T_b}, \tag{4}$$

or

$$E = 1 - \frac{T_a}{T_b}\frac{\frac{T_d}{T_a} - 1}{\frac{T_c}{T_b} - 1}. \tag{5}$$

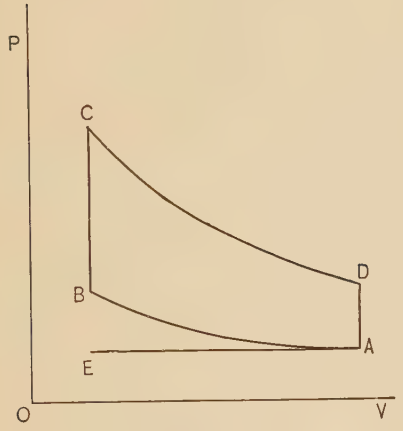


Fig. 15.

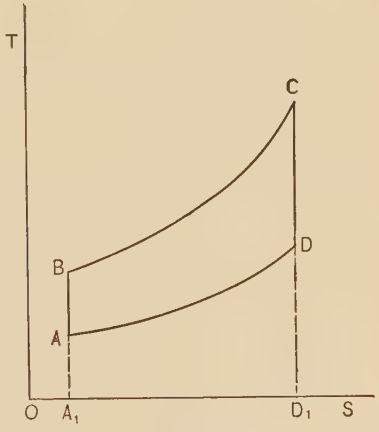


Fig. 16.

Otto Diagram.

From **adiabatic** relations we get

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{n-1},$$

$$T_c = \left(\frac{V_d}{V_c}\right)^{n-1} T_d,$$

$$T_b = \left(\frac{V_a}{V_b}\right)^{n-1} T_a.$$

But as $V_b = V_c$ and $V_a = V_d$ $\therefore$ $\frac{V_a}{V_b} = \frac{V_d}{V_c}$ and $\frac{T_c}{T_b} = \frac{T_d}{T_a}$.

Then

$$E = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} = 1 - \frac{1}{r^{n-1}}, \tag{6}$$

where $r = \frac{V_a}{V_b}$, the ratio of compression.

From equation (1) we get

$$\frac{T_c}{T_b} = \frac{Q_{bc}}{c_v T_b} + 1,$$

or

$$\frac{T_c}{T_a} = r^{n-1}\left(\frac{Q_{bc}}{c_v T_b} + 1\right), \tag{7}$$

which gives us the range of temperatures in the cycle.

Comparison of Cycles. — Of the five cycles analyzed, two are not used in practice. The **Carnot** is a **theoretical** cycle used only as a standard. To use this cycle and obtain its theoretical efficiency would require higher pressures and temperatures than could be handled in actual practice.

The efficiency of the **Lenoir** cycle, due to absence of compression, is so low that this cycle was discarded after a few years of use and has never been taken up again.

The **Brayton, Diesel** and **Otto** cycles are shown, superimposed, in Fig. 17. The **minimum temperature** at A and the **maximum temperature** at C are the **same** for all three. It is seen, then, that the **Brayton** cycle $AB'CD'$ has the largest area and the **Otto** cycle $ABCD$ the smallest. Hence, between the same temperature limits and the same maximum pressure P_c the **Brayton** cycle is the most efficient, and the **Otto** the least efficient. Comparing the maximum volumes, it is seen that the **Otto** and **Diesel** cycles have the same maximum volumes V_d while the **Brayton** cycle requires a greater volume, as shown by the point D'. The **Diesel** cycle, therefore, combines the advantages of the high efficiency of the **Brayton** cycle due to high compression pressure and the smaller cylinder volume of the **Otto** cycle.

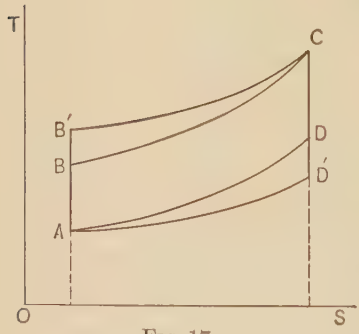


Fig. 17.

Further practical comparisons of the **Otto** and **Diesel** cycles will be discussed when the design of these two types is taken up later.

CHAPTER IV

FUELS AND COMBUSTION

Combustion — Air Required — Gases — Hydrocarbons — Calorimeters — Specific Heat — Vapor Pressure

General. — **Carbon** and **hydrogen** are the elements that form combinations to furnish heat in gas-engine fuels. Some fuels contain, in combination with hydrogen and carbon, small percentages of oxygen. These fuels are usually artificial, such as producer gas and coal gas. When oxygen is present in coal, it is looked upon as a detriment, as it is already combined with hydrogen, and reduces the total heating value of the hydrogen content.

The principal **natural fuels** that are used for internal combustion engines are **coal, wood, peat, lignite, oil** and **natural gas.** Among the **artificial fuels** are **illuminating gas, blast-furnace gas, producer gas, coke-oven gas, alcohol,** etc. **Producer gas** plays a very important part in gas-engine work. Indications seem to point to producer gas as the solution to the gas-engine problem, since, by the producer alone, can the low-grade fuels be made available for furnishing power efficiently.

Combustion. — The process of **combustion** is nothing more than the union of a fuel with oxygen to form an oxide. Slow and gradual union takes place, but is not brought under the term **combustion** as no sensible heat is developed. Thus, the rusting of iron is a union with oxygen, as is the decaying of animal and vegetable matter, but these chemical reactions do not come under the term combustion in the sense in which we wish to use it in connection with this work.

In a **furnace** or **stove** air is brought into contact with coal (carbon) at a high temperature. The oxygen of the air unites with the hot carbon and, if sufficient air be supplied, carbon dioxide results. If the air or oxygen supply be limited, the resulting product of combustion is mainly carbon monoxide. If the carbon burns to carbon dioxide, a certain amount of heat is generated, or liberated, for each pound of carbon consumed. If the air supply

is scant and the carbon burns to the monoxide, then a much smaller amount of heat will be generated. This, in general, illustrates perfect and imperfect combustion.

Combustion of Carbon. — The questions of finding heating values of gaseous mixtures and air required per unit, by weight or volume, may best be solved by use of the chemical or combustion formulæ. Thus, when carbon burns completely, we have

$$\begin{array}{ccccc} C & + & O_2 & = & CO_2 \\ 12 & & 32 & & 44 \end{array}$$

The numbers refer to the atomic weights of the elements, and they show that 12 parts of carbon, **by weight,** unite with 32 parts of oxygen to form 44 parts of carbon dioxide (CO_2). Thus, for each pound of carbon burned to CO_2, there are required $\frac{3}{1}\frac{2}{2} = 2.67$ pounds of oxygen. The heat evolved during the burning of 1 pound of carbon is 14,600 British thermal units (B.t.u.).

For incomplete combustion we have

$$\begin{array}{ccccc} 2\,C & + & O_2 & = & 2\,CO \\ 24 & & 32 & & 56 \end{array}$$

Thus 24 parts of carbon unite with 32 parts of oxygen forming 56 parts of carbon monoxide (CO), or 1 pound of carbon requires 1.33 pounds of oxygen to burn to 2.33 pounds of CO. The heat generated in the partial combustion of 1 pound of carbon to CO is 4380 B.t.u. The difference 14,600 − 4380 = 10,220 is still contained in the 2.33 pounds of CO. If this CO is lost, the heat, of course, is lost. If the CO is burned, the heat is regained. Thus we would expect the heating value of the CO to be $\frac{10{,}220}{2.33} =$ 4380 B.t.u. per pound, which is the figure usually given.

The equation for the combustion of the CO is

$$\begin{array}{ccccc} 2\,CO & + & O_2 & = & 2\,CO_2 \\ 56 & & 32 & & 88 \end{array}$$

which indicates that 1 pound of CO requires 0.572 pound of oxygen for combustion, resulting in 1.572 pounds of CO_2.

In the pound of CO there will be $\frac{1}{2}\frac{2}{8}$ pound of carbon. This carbon requires 0.572 pound of oxygen to complete the combustion or $\frac{0.572}{12} \times 28 = 1.33$ pounds of oxygen per pound of carbon in the CO. Thus it requires the same amount of oxygen to burn

the carbon to CO and then to CO_2 as to burn the carbon directly to CO_2 in one operation.

Combustion of Hydrogen. — When hydrogen is burned we have the chemical reaction

$$\begin{array}{ccccc} 2\,H_2 & + & O_2 & = & 2\,H_2O \\ 4 & & 32 & & 36 \end{array}$$

showing that 1 pound of hydrogen unites with 8 pounds of oxygen, forming 9 pounds of water.

When this process is reversed and steam is forced through a bed of incandescent carbon, the opposite reaction results. The steam is broken up into hydrogen and oxygen and the latter unites with the carbon in the coke forming carbon monoxide.

For that reaction the equation is

$$\begin{array}{ccccccc} H_2O & + & C & = & CO & + & H_2 \\ 18 & & 12 & & 28 & & 2 \end{array}$$

showing that 1 pound of water unites with 0.667 pound of carbon, resulting in 1.556 pounds of carbon monoxide and 0.111 pound of hydrogen.

When hydrogen is burned to water at 32° F., the heat generated per pound of hydrogen is 62,100 B.t.u. When the reverse process is used, that is, when water is dissociated, for every pound of hydrogen that results 62,100 B.t.u. must be supplied. Since it requires 9 pounds of water to be dissociated to form 1 pound of hydrogen, the heat necessary to dissociate 1 pound of water is 6900 B.t.u.

Combustion of Hydrocarbons. — Some of the gas-engine fuels, natural and illuminating gas and the producer gas made from bituminous coal, contain **heavy hydrocarbons.** The principal ones of these are

Methane or **marsh gas** CH_4.

Ethylene or **olefiant gas** C_2H_4.

Both these gases burn to carbon dioxide and water and their chemical equations may be written as for carbon and hydrogen alone. For methane we have

$$\begin{array}{ccccccc} CH_4 & + & 2\,O_2 & = & CO_2 & + & 2\,H_2O \\ 16 & & 64 & & 44 & & 36 \end{array}$$

It is evident that 1 pound of methane requires 4 pounds of oxygen for combustion, resulting in $2\frac{3}{4}$ pounds of carbon dioxide

and $2\frac{1}{4}$ pounds of water. The heat generated in the combustion of 1 pound of the gas is 24,000 B.t.u.

When ethylene is burned the equation is

$$\underset{28}{C_2H_4} + 3\,\underset{96}{O_2} = 2\,\underset{88}{CO_2} + 2\,\underset{36}{H_2O}$$

indicating that 1 pound of C_2H_4 requires 3.42 pounds of oxygen, forming 3.14 pounds of CO_2 and 1.28 pounds of water. The heat generated is about 21,500 B.t.u. per pound.

A **general equation** for the oxygen required for the combustion of any hydrocarbon may be written:

$$2.667\,C + 8\,H,$$

where C and H are the proportions of carbon and hydrogen respectively entering into the composition.

Atmospheric Air. — In all forms of internal combustion engines the oxygen required for the combustion of the fuel is obtained from the atmosphere. Since air contains only a small proportion of oxygen, we must carry our calculations one step further and find the quantity of air required for combustion of the various fuels.

Air is composed principally of oxygen and nitrogen in the following proportions:

By weight		By volume	
Oxygen	0.231	Oxygen	0.21
Nitrogen	0.769	Nitrogen	0.79
	1.000		1.00

The other constituents of the atmosphere, CO_2, water vapor, etc., do not form a large enough proportion to have any influence on gas-engine calculations.

When the weight of oxygen required for combustion has been found, only a short calculation is required to find the weight of air corresponding. Thus, for the complete combustion of one pound of carbon, 2.67 pounds of oxygen are required. The air necessary to furnish this oxygen, since air contains 23 per cent oxygen by weight, will be $\frac{2.67}{0.23} = 11.6$ pounds.

For the combustion of hydrogen, 8 pounds of oxygen are required, necessitating the use of $8 \div 0.23 = 34.63$ pounds of air.

Then for the complete combustion of methane which is 75 per cent carbon and 25 per cent hydrogen, there would be required

$$0.75 \times 11.6 + 0.25 \times 34.63 = 17.36 \text{ pounds of air.}$$

When fuels, such as alcohol, contain oxygen, this oxygen may be subtracted from the theoretical oxygen required, as the oxygen in the fuel unites with some of the hydrogen.

Volume of Air Required for Combustion. — The volume of 1 pound of air at 32° F. is 12.39 cubic feet and at 62° F. is 13.14 cubic feet. After the weight of air has been determined, the volume at 32 degrees or 62 degrees may be found by multiplying by 12.39 or 13.14. Thus, the 11.6 pounds of air required to burn a pound of carbon to CO_2 would occupy a space of 143.7 cubic feet at 32 degrees, atmospheric pressure, or 152.4 cubic feet at 62 degrees. It would be possible at this point to derive formulæ showing the volume of air at any temperature required for any fuel containing C, H_2 and O_2. These formulæ, however, are easily forgotten if not used constantly, so that reference must be made to some book containing the formulæ every time they are to be used. The author prefers to use the combustion equations each time that the air required for a fuel is to be calculated. In that way bulky formulæ are done away with, and the chance for errors to creep in, due to the misuse of these formulæ, is very slight.

Air Required for a Compound Gas. — The air required for the combustion of a pound of compound gas may be found in the same way that it was found for methane and ethylene. If no tables are at hand, the combustion equation may be written for each constituent and the oxygen required **per pound** of this constituent found. If this value is then multiplied by the percentage by weight of that constituent in the mixture, and these products added, the total will be the **oxygen** required per pound of the compound. This can easily be changed into pounds or cubic feet of **air.**

Example: Find the pounds of air required per pound of blast-furnace gas of the following composition:

	Per cent
CO	27
H_2	3
CO_2	12
N_2	58
	100

In order to use the chemical equations, this analysis must be reduced to the weight basis. If each percentage, divided by 100, is multiplied by the molecular density of the corresponding gas, the results will give the weight of each constituent compared to the weight of a molecule of hydrogen. If these individual weights be divided by their total, the quotient will be the percentage weight of that constituent.

$CO = 0.27 \times 14 =$	3.78	25.90
$H_2 = 0.03 \times 1 =$	0.03	00.21
$CO_2 = 0.12 \times 22 =$	2.64	18.10
$N_2 = 0.58 \times 14 =$	8.12	55.79
	14.57	100.00

Since the CO and the hydrogen are the only combustibles in this gas, we have only to find the air required for them. We have already found that 1 pound of CO requires 0.572 pound of oxygen for combustion and that 1 pound of hydrogen requires 8 pounds of oxygen. The total oxygen per pound of this gas will be 0.165 pound.

0.572×0.259	$= 0.148$
8×0.0021	$= 0.017$
	0.165

The air to correspond will be $\frac{0.165}{0.23} = 0.72$ pound.

Air Per Cubic Foot of Gas. — In gas-engine design it is frequently necessary to find the volume of air required per cubic foot of gas. If the density of the gas is known, the relative volumes of air and gas may be found from the relative weights found in the preceding article. A more convenient method is to use a table giving the cubic feet of air required per cubic foot of gas for the different fuel gases. Such tables are given here, Tables I and II. Using Table II we find that 1 cubic foot of CO requires 2.4 cubic feet of air for combustion and a cubic foot of hydrogen the same. Using the volumetric analysis, we find the air required per cubic foot of the gas mentioned above is 0.72 cubic foot.

$CO\ 0.27 \times 2.4$	$= 0.648$
$H_2\ 0.03 \times 2.4$	$= 0.072$
	0.720

Volume of Products of Combustion. — It is sometimes of importance to find the volume of products of combustion of a gas.

The weight of the products of combustion of the fuel is 1 pound, plus the weight of air supplied per pound of fuel.

TABLE I

DATA ON COMMON FUELS, ETC.

	Chemical symbol.	Molecular weight.	Weight per cu. ft. at 14.7 lbs. pressure and		Calorific value.					
					B.t.u. per lb.		B.t.u. per cu. ft. of gas.			
							High value at		Low value at	
			32° F.	62° F.	High.	Low.	32° F.	62° F.	32° F.	62° F.
Hydrogen	H_2	2	0.005591	0.005269	62,100	52,275	347	327	292	275
Methane (marsh gas)	CH_4	16	0.04473	0.04215	24,000	21,544	1073	1013	968	910
Ethylene (olefiant gas)	C_2H_4	28	0.07827	0.07376	21,900	20,496	1714	1615	1600	1512
Acetylene	C_2H_2	26	0.07268	0.06849	21,860	21,094	1592	1500	1540	1440
Ethane	C_2H_6	30	0.08386	0.07903	22,340	20,375	1880	1800	1700	1620
Butylene	C_4H_8	56	0.15655	0.1476	20,860	19,405	3264	3080	3036	2860
Benzene	C_6H_6	78	0.21805	0.2055	18,100	17,343	3940	3720	3780	3560
Carbon monoxide	CO	28	0.07827	0.07376	4,380	4,380	344	324	344	324
Carbon	C	12	.....	.....	14,600	14,600	....	....	....	....
Carbon dioxide	CO_2	44	0.123	0.1159	.....	.....	....	....	....	....
Nitrogen	N_2	28	0.07827	0.07376	.....	.....	....	....	....	....
Oxygen	O_2	32	0.08936	0.08421	.....	.....	....	....	....	....

TABLE II

DATA ON COMMON FUELS

	Required weight of air per lb. of fuel, lbs.	Weight of combustion products per lb. of gas.			Volume of combustion products per lb. of gas at 32° F.			Cu. ft. of air required for combustion.	
		CO_2, lbs.	H_2O, lbs.	N_2 lbs.	CO_2 cu. ft.	H_2O cu. ft.	N_2 cu. ft.	Per lb. of gas at 62° F.	Per cu. ft. gas.
Hydrogen	34.64	....	9	26.64	.....	178.87	340.46	456	2.4
Methane (marsh gas)	17.32	2.75	2.25	13.32	22.36	44.72	170.23	228	9.61
Ethylene (olefiant gas)	14.85	3.14	1.29	11.42	25.55	25.58	145.91	195	14.3
Acetylene	13.32	3.39	0.69	10.24	27.52	13.75	130.94	175	12.0
Ethane	16.16	2.93	1.80	12.43	23.85	35.77	158.87	213	16.83
Butylene	14.85	3.14	1.29	11.42	25.55	25.58	145.91	195	28.8
Benzene	13.32	3.39	0.69	10.24	27.52	13.75	130.94	175	36.0
Carbon monoxide	2.48	1.57	....	1.91	12.78	.....	24.35	33	2.4
Carbon	11.56	3.67	....	8.89	29.81	.....	113.59	152	.....

To determine the total volume of the combustion products some standard of temperature and pressure must be decided on. If we select 32° F. and atmospheric pressure, then a pound of CO_2 will occupy 8.13 and a pound of nitrogen 12.78 cubic feet. The H_2O

vapor formed by the combustion of hydrogen would condense if the temperature were lowered to 32° F. at atmospheric pressure. If we consider this vapor to be a gas at that temperature, then the volume of a pound would be 19.27 cubic feet.

Using these values we will proceed to find the volume of the products of combustion of 1 pound of methane CH_4. From the preceding articles we found that a pound of carbon requires 2.67 pounds of oxygen for combustion and a pound of hydrogen 8 pounds of oxygen. Also, from the molecular weight it is evident that methane is composed of 75 per cent carbon and 25 per cent hydrogen by weight. The volume of the combustion products will then be

$$0.75\ (1 + 2.67)\ \ 8.13 = 22.38 \text{ for the } CO_2,$$
$$0.25\ (1 + 8.0\ )\ 19.27 = 43.36 \text{ for the } H_2O,$$

$(0.75 \times 2.67 + 0.25 \times 8.0\ )\ 12.78 \times 3.33^* = 170.34$ cubic feet at 62° F. for the nitrogen.

It is often necessary to find the volume of the exhaust gases at a higher temperature than the standard, 32 degrees. To do this we may use the law that, at constant pressure, the volume of the gas varies as the absolute temperature. Then the volume at any temperature "t" would be:

$$\text{Vol. at "}t\text{" °F.} = \frac{460 + t}{460 + 32} \times \text{vol. at } 32° \text{ F.}$$

At any temperature below 212 degrees, when the pressure is atmospheric, the volume of the water vapor would contract to 0.0016 of the volume computed for 32 degrees, so that for any temperature below the boiling point we may assume the volume of the water vapor to be zero.

Heating Value of Fuels Containing Hydrogen. — We have seen that 1 pound of hydrogen unites with 8 pounds of oxygen, forming 9 pounds of water, and generates 62,100 heat units in forming the combination. This heating value is found in a calorimeter by burning the gas at constant pressure. The heat is absorbed by the products of combustion as sensible heat and by the water vapor as latent heat. To find the heating value at a standard of 32 degrees, the products of combustion are cooled to that point, and as that point is far below the boiling point of water at

* This factor shows the relation of nitrogen to oxygen in the air; there are 3.33 pounds of nitrogen present for every pound of oxygen.

atmospheric pressure, the latent heat of the vapor is given up and is measured as a part of the 62,100 B.t.u.

If, in using the calorimeter, we did not cool the products of combustion below 212 degrees, the presence of the latent heat would not become manifest and the result would be a lower heating value determined per pound of hydrogen than if the products of combustion were cooled below 212 degrees. Similarly, when the heat of combustion is used for heating the charge in a gas-engine cylinder, the total heat of the hydrogen is generated, but if the exhaust leaves the engine at a higher temperature than 212 degrees, as it always does, the latent heat of the water vapor is carried with it and has had no part in producing work in the cylinder. When the water vapor formed from the hydrogen in the fuel is not condensed the heat generated per unit of fuel is termed the lower heating value.

The question of which heating value to charge up to a gas-engine in finding the thermal efficiency has often been raised. It seems fair to charge the engine with the higher heating value, even though the engine does not use all of it. The latent heat of the steam is charged against the steam engine, so should the latent heat of the water vapor be charged against the gas engine.

In some instances, in studying the effects of heat on the charge in the engine cylinder, it will be necessary to use the lower heating value. This question will arise in determining the size of the cylinder to generate a given power.

Any gas fuel containing hydrogen will have a **higher** and a **lower** heating value, and the latter is always found by subtracting from the former the latent heat in the steam formed by the combustion of hydrogen in the fuel. If we take the standard at 32 degrees, we must subtract from the higher heating value the latent heat of the steam at 32 degrees. This may be obtained from Regnault's formula,

$$H = 1091.7 - 0.695\,(t - 32);$$

or at 32 degrees this becomes 1091.7. As 1 pound of hydrogen forms 9 pounds of water at combustion, we must subtract $9 \times 1091.7 = 9825$ from 62,100, giving 52,275 as the lower heating value.

If, in any hydrocarbon, the ratio of hydrogen expressed in decimals be x, the vapor formed at combustion of the gas will be $x \times 9$

pounds per pound of gas, and the heat of vaporization will be $x \times 9 \times 1091.7$. Therefore, $x \times 9825$ heat units should be subtracted from the higher calorific value of the gas in order to get the lower.

The Gas Calorimeter. — Calorimeters, in general, are arranged for burning in the apparatus a known quantity of fuel with oxygen or air for combustion, and absorbing the heat by means of a measured quantity of water. If the increase in temperature of water is observed, the heat generated per pound of fuel will be obtained from the equation

$$Q = \frac{W(T_2 - T_1)}{L},$$

where W is the weight of the cooling water, $T_2 - T_1$ its increase in temperature and L the weight of the fuel burned.

The Junker type calorimeter is one of the best known for finding the heating value of gases. The instrument shown in Fig. 18 is of this type and is manufactured by the D. McDonald & Co. Works of the American Meter Co.

The gas under test is metered in the wet test meter shown at the left. This meter is equipped with a pressure gauge, a thermometer, and a bell which strikes at each revolution of the large pointer. From the meter the gas passes through a pressure regulator shown between the meter and the burner. The burner, which is shown withdrawn from the calorimeter proper, is of the Bunsen type, and is equipped with a micrometer cock. Three adjustable baffle plates are supplied to prevent downward radiation and consequent loss of heat from the combustion chamber.

The calorimeter proper consists of a central combustion chamber around which is located the water heating element consisting of 40 tinned copper tubes arranged in two staggered concentric rows. The hot gases leaving the burner pass up through the combustion chamber, down through the water heating element, and out of the flue shown at the bottom on the right side.

The water enters the calorimeter through an inlet weir at the left, and passes out through a similar weir at the right. The function of these weirs is to maintain a uniform head of water regardless of any variation of depth in the supply tank. The flow is adjusted through a micrometer cock on the inlet, and is passed to the weighing bucket or diverted to the waste through a three-way cock on the outlet. The temperatures of the inlet

and outlet water are read by means of the two thermometers shown at the top of the calorimeter. These are graduated to one-tenth of a degree, and are equipped with microscopic sights.

Two copper pails, evenly matched in weight, and precision scales are furnished for accurately weighing the water.

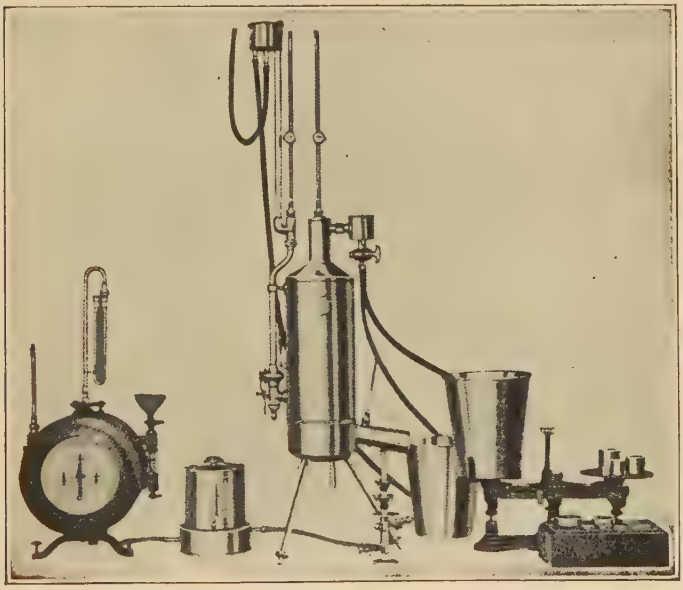


Fig. 18.—The Junker Gas Calorimeter.

The Mahler Bomb Calorimeter.—For solid or liquid fuels the **Mahler bomb calorimeter** is frequently used. It consists of a small steel vessel or bomb lined, usually, with an enamel on which acids will have no effect. In this bomb a small known weight of fuel is burned in an atmosphere of dense oxygen, while the instrument is submerged in a known quantity of water which absorbs the heat of combustion.

On the cap of the bomb is suspended a platinum pan on which is placed the fuel to be tested. If it is coal, it must be finely powdered to insure complete combustion. After the coal and platinum pan are placed in position, a fine iron fuse wire is attached to two terminals in the cap, and passed through the sample of coal. After the coal and fuse wire are in place, suspended from

the cap, this cap is screwed tightly on the bomb, leakage being prevented by a lead gasket. The bomb is then charged with about 20 atmospheres of oxygen, placed in the can of carefully weighed water and the firing wires are attached to the terminals. The can containing the water is insulated so that little or no heat will be transmitted to the outer air. In order to measure the heat thus transmitted, the can and insulation are placed in a second receptacle containing water. By measuring the rise in temperature of this water, the radiation may be found.

Since the sample of coal used is small, about 0.6 gram, and the water in which the bomb is submerged is usually about 2000 grams, the rise in temperature will be very small, from 1.5° to 3° C. Very sensitive thermometers must be used to measure the rise, and readings must be carefully taken.

While this is just an outline of the procedure when a bomb calorimeter is used, a more thorough explanation would be burdensome unless the reader is especially interested. If tests are to be made with such a calorimeter, books on fuel analysis should be consulted for methods in computing radiation, corrections for specific heat of water, sulphur in coal, heating of fuse wire, etc.

Heating Value of Fuel by Formula. — In many cases the chemical analysis of a fuel is obtained more easily than a calorimeter test, and the heating value of a solid fuel may be approximated very closely by use of this analysis.

One well-known formula used for this work is credited to Dulong:

$$Q = 14{,}600\,\mathrm{C} + 62{,}100\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right), \qquad (1)$$

where Q is the heating value of the fuel, B.t.u. per pound, C the percentage carbon, H the percentage hydrogen and O the percentage oxygen. C, H and O are to be expressed in decimal parts of the total weight of the fuel as a unit.

This formula is based on two important laws by Dulong:

"The heating value of a compound fuel is the sum of the heating values that would be obtained by the combustion of each component separately, providing that the fuel does not contain oxygen and hydrogen chemically combined with it."

"When oxygen and hydrogen do exist in chemical combination with a fuel, it is only the part of the hydrogen outside of that required for form-

ing water with the oxygen that will evolve heat; and the oxygen required for combustion will be reduced from the amount the analysis would show by the amount of oxygen in the fuel."

For liquid hydrocarbons the following approximate formula seems to give results which check the calorimeter tests closely, and consequently it is often used.

$$Q = 14{,}600\,\mathrm{C} + 52{,}230\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right). \qquad (2)$$

If formula (1) be applied to methane, for instance, the result will be

C	$= 0.75 \times 14{,}600$	$= 10{,}950$
H_2	$= 0.25 \times 62{,}100$	$= 15{,}500$
	Total	$26{,}450$

and by formula (2) we get

C	$= 0.75 \times 14{,}600$	$= 10{,}950$
H_2	$= 0.25 \times 52{,}230$	$= 13{,}058$
	Total	$24{,}008$

The first formula gives a result that is much higher and the second a result that checks the heating value given for methane in Table I. This difference is due to the fact that the heat to break up the chemical union of the carbon and hydrogen is not available, that the actual heat evolved is the theoretical heating value less the heat of the chemical union; hence the discrepancy in the results found above.

Specific Heat and Flame Temperature. — The specific heat of a substance has already been defined as the quantity of heat necessary to raise the temperature of a unit weight of that substance one degree, the specific heat of water being unity.

Gases have two specific heats. If a gas is heated at constant volume, no external work is done and consequently no heat need be supplied to do this work. If heat is added while pressure is constant, then the volume increases, work is done and heat must be added to do this work. Therefore the specific heat at constant pressure is greater than the specific heat at constant volume.

The specific heats of some of the more important gases are given in Table III. These are the values found by Regnault.

The maximum temperature of the products of combustion of a fuel, or, in other words, the **flame temperature,** may be approxi-

TABLE III

SPECIFIC HEATS OF GASES AT 212° FAHRENHEIT

	At constant pressure, c_p.	At constant volume, c_v.	$\frac{c_p}{c_v}$.
Air	0.2375	0.1700	1.41
Superheated steam H_2O	0.4805—	0.3585	1.34
Hydrogen H_2	3.4090	2.4177	1.41
Oxygen O_2	0.2175	0.1543	1.41
Nitrogen N_2	0.2438	0.1729	1.41
Carbon monoxide CO	0.2479—	0.1738	1.41
Carbon dioxide CO_2	0.2169	0.1570	1.29
Marsh gas CH_4	0.5929	0.4505	1.316
Olefiant gas C_2H_4	0.4040	0.3200	1.26
Average anthracite gas	0.24 to 0.26	.....	.....
Average bituminous gas	0.24 to 0.26	.....	.....
Average blast-furnace gas	0.23	.....	.....
Average chimney gas	0.24	.....	.....

mated by assuming that all the heat of combustion is used to heat the products of combustion, and that the specific heat of the products is constant. Then the temperature rise resulting from the combustion would be found by the following formula:

$$T = \frac{Q}{CW},$$

where Q is the lower heating value of the fuel per pound, C is the mean specific heat of the products of combustion and W is the weight of the products of combustion per pound of the fuel. The actual temperature after combustion will be the sum of the original and the rise.

Example: Find the flame temperature of the blast-furnace gas whose composition by weight is CO = 22 per cent, H_2 = 1 per cent, CO_2 = 14 per cent and N_2 = 63 per cent. Excess air is 15 per cent. Combustion is to take place at atmospheric pressure.

The first step is to find the air required for perfect combustion. This is found by the help of Table II.

$$\begin{aligned} \text{Air for CO} &= 0.22 \times 2.477 = 0.5460 \text{ pound} \\ \text{Air for } H_2 &= 0.01 \times 34.64 = 0.3464 \text{ pound} \\ \text{Total} &\quad 0.8924 \text{ pound} \\ \text{Excess} &= 0.15 \times 0.8924 = 0.1340 \text{ pound.} \end{aligned}$$

Total weight of combustion products will be

$$0.8924 + 0.1340 + 1.000 = 2.0264.$$

The weight of the products of combustion, per pound of gas, may also be found from the same table.

	Weights of products of combustion.			
	H_2O	CO_2	N_2	Air.
From CO, 0.22 pound	.....	0.348	0.419	.....
From hydrogen, 0.01 pound	0.09	.....	0.267	.....
From CO_2	.....	0.140	.....	.....
From nitrogen	.....	.....	0.630	.....
From excess air	.....	.....	.....	0.134
Totals	0.09	0.488	1.316	0.134

To find the specific heat of the mixture, each component must be multiplied by its specific heat, the products added and divided by the total weight. For this gas we have

H_2O	$0.09 \times 0.481 =$	0.0433
CO_2	$0.488 \times 0.217 =$	0.1059
N_2	$1.316 \times 0.244 =$	0.3135
Air	$0.134 \times 0.238 =$	0.0319
	2.028	0.4946

This value 2.028 should check the value 2.0264 found above.

The mean specific heat of the combustion products will be:

$$\frac{0.4946}{2.028} = 0.245.$$

If the heating value of the fuel is 1486 B.t.u. per pound, then the theoretical temperature of the flame will be

$$\frac{1486}{2.028 \times 0.245} = 2990° \text{ F. above the atmospheric temperature.}$$

If the gas is burned at constant volume, the specific heat at constant volume would have to be used to find the flame temperature. This is found to be .175 for the gas chosen by the same method used above forgetting the value at constant pressure, and as it is lower than the specific heat at constant pressure, the corresponding temperature will be higher, or

$$T_v = \frac{1486}{0.175 \times 2.028} = 4186 \text{ degrees}$$

above atmospheric temperature. Since, in the engine cylinder, not all the heat of combustion is utilized in raising the temperature of the products of combustion, this temperature is never approached in practice.

Density of Gases. — Avogadro's law states the fact that equal volumes of all gases at the same temperature and pressure contain the same number of molecules, and as a consequence it follows that the ratio between the weights of equal volumes, or between the weights of unit volumes, of any two gases is as the ratio of their molecular weights.

The molecular weights of substances are usually referred to the weight of the hydrogen molecule as a unit. Thus, since the molecular weight of hydrogen is 2, if we call that of any other gas m, the ratio of the weight of unit volumes of hydrogen and the given gas will be as 2 to m or as 1 to $\frac{m}{2}$. Hence, the weight of a unit volume of a gas, or the density, is

$$\text{density of a gas} = \frac{m}{2} \times \text{density of hydrogen.}$$

To determine the weight per cubic foot of any gas it will only be necessary to multiply the density of hydrogen by half the molecular weight of the gas in question. The weight per cubic foot of hydrogen is, at 32° F. and 14.7 pounds (atmospheric) pressure, 0.005591 pound. At the same standard conditions the number of cubic feet in a pound of hydrogen will be the reciprocal of 0.005591 or 178.87.

As an example, the weight per cubic foot of marsh gas CH_4 will be $\frac{16}{2} \times 0.005591 = 0.044728$.

The density of a gas at constant pressure varies inversely as the absolute temperature. Thus at t degrees the density will be

$$D_t = \frac{459.2 + 32}{459.2 + t^\circ} D_{32}.$$

It may sometimes be more convenient to refer the density of a gas to the density of air instead of hydrogen. As dry air at a temperature of 32° F. and a pressure of 14.7 pounds is $\frac{0.08728}{0.005591}$ = 14.44 times the weight of the hydrogen, then density of a gas referred to air = $\frac{m}{28.88}$ where m is the molecular weight of the gas.

The volume per pound of gas is the reciprocal of the density and since we have

$$\text{density of a gas} = \frac{m}{2} \times 0.005591,$$

then

$$\frac{1}{\text{density of gas}} = \frac{2}{0.005591\ m} = \frac{357.74}{m}.$$

That is, to find the cubic feet per pound of a given gas, divide 357.74 by the molecular weight of that gas.

Heating Value Per Unit Volume. — It is usually more convenient to measure the fuel supply of a gas engine in cubic feet than in pounds. Then, instead of using the heating value per pound of the gas, it will be necessary to use the heating value per cubic foot. If the heating value per pound be known, the heating value per cubic foot may be found by multiplying the B.t.u. per pound by the density of the gas.

To reduce the volume of gas to a standard condition, usually 32 degrees and atmospheric pressure, the following formulæ will be convenient. The subscript x will be used to designate any condition and s the standard condition.

T_x = any absolute temperature = t_x + 459.2° F.,
P_x = any absolute pressure = 14.7 + gauge pressure,
V_x = corresponding volume, in cubic feet,
T_s = standard absolute temperature = 491.2° F.,
P_s = standard pressure = 14.7 pounds per square inch,
V_s = volume of gas at standard temperature and pressure,

then

$$V_s = V_x \frac{P_x}{P_s} \frac{T_s}{T_x} = V_x \frac{491.2\ P_x}{14.7\ T_x}.$$

It is apparent that the denser the gas, the greater its heating value per cubic foot, or the heating value varies directly with the pressure and inversely as the temperature. Thus, if the heating value of hydrogen be 347 B.t.u. per cubic foot at 32° F. and atmospheric pressure, its heating value at 62 degrees and the same pressure will be $347 \times \frac{491.2}{521.2} = 327$ B.t.u.

At 212° F. and 14 pounds pressure, absolute, the heating value would be $347 \times \frac{491.2}{671.2} \times \frac{14.0}{14.7} = 241.5$ B.t.u.

Vapor Pressure. — If the laws of perfect gases hold with fuel vapors, then the weight of fuel vapor present in a given volume, at a fixed temperature, is proportional to its vapor pressure. There is a definite limit to the amount of condensable vapor that can

exist in a given space, at any given temperature. There is, for each temperature of the vapor, a limit to the pressure that it can sustain without becoming partially condensed back into the liquid.

In raising the temperature of a vapor its maximum vapor pressure, or saturation pressure, is increased and for each pressure there is a certain temperature where partial condensation begins. This temperature is called the saturation temperature at that given pressure. If the temperature of a saturated vapor is lowered, then the vapor becomes moist, and if its temperature is raised, it becomes superheated.

In Table IV is given the vapor pressures of saturation for different motor fuels and water. The values for the alcohols are from the "Fifth Revised Edition of the Smithsonian Physical Tables" and those for benzol, hexane and gasoline from Sorel's "Carburation et Combustion dans les Moteur a Alcool."

When two gases are enclosed in the same receptacle and subjected to an external pressure, both help to sustain that pressure in the ratio of their respective vapor pressures, and the sum of the vapor pressures of the two gases must equal the pressure to which they are subjected.

TABLE IV

VAPOR PRESSURE OF SATURATION

Temperature.		Vapor pressure of saturation in millimeters of mercury.						
							Gasoline.	
Degs. C.	Degs. F.	Pure ethyl alcohol C_2H_6O.	Pure methyl alcohol CH_4O.	Benzol.	Water.	Hexane C_6H_{14}.	Automobiline. Sp. density at 62° F., 0.703.	Stelline. Sp. density at 62° F., 0.673.
0	32	12.7	27	27	5	45	99	164
5	41	17.6	37	36	7	58	115	190
10	50	24.2	50	45	9	74	133	220
15	59	33.0	67	61	13	95	154	255
20	68	44.5	89	77	17	119	179	296
25	77	59.4	116	96	24	154	210	358
30	86	78.5	150	120	32	184	251	433
35	95	102.9	192	156	42	228	301	512
40	104	133.7	244	188	55	276	360	596
45	113	172.2	306	224	71	335	422	685
50	122	219.9	382	271	92	401	493	792
55	131	278.6	472	326	117	482	561	.....
60	140	350.2	580	390	149	567	648	.....
65	149	436.9	707	468	187	674	739	.....

In a saturated mixture of fuel vapor and air, if we assume it to follow the laws of perfect gases, the ratio of volumes of vapor and air present in a given volume equals the direct ratio of their vapor pressures.

When the vapor pressure of a fuel is, under ordinary atmospheric conditions, much weaker than that of air, it may happen that the proportion of fuel to air is too small to form an explosive mixture, or the fuel mixture is too weak to be explosive. This is true of alcohol and kerosene and shows why these fuels must be heated before their vapor pressure is high enough to form an explosive mixture. With lighter fuels, such as benzine and gasoline, this is not the case. Their vapor pressures are so high that a saturated mixture under atmospheric conditions may be too rich to explode and must be diluted. This point will be taken up later when the common fuels are discussed further.

EXERCISES

1. A fuel contains by weight:

	Per cent
H_2	30
CO	37
N_2	Balance

Find higher heating value per pound and weight of air required for complete combustion.

2. Find lower heating of same.

3. Given a fuel containing by weight:

	Per cent
H_2	8
CO	43
CH_4	5
C_2H_4	7
CO_2	14
N_2	Balance

Find volume of air required per pound of gas.

4. Find the volume of combustion products per pound of the gas given in Prob. 3.

5. Given an analysis by volume as follows:

	Per cent
CO_2	13
H_2	3
CH_4	6
CO	22
N_2	56

Find analysis by weight.

6. (*a*) Use the Dulong formula to find the heating value of coal containing 64 per cent C, 5 per cent H_2 and 3.2 per cent O_2. (*b*) Use it for finding the heating value of methane and ethylene and check these values by Table I.

7. Find the flame temperature of gas in Prob. 5, gas to be burned at constant pressure. No excess air.

8. Find the heating value per cubic foot of gas in Prob. 5 when the gas temperature and pressure are 100° F. and 5 lbs. per sq. in. gauge, respectively.

CHAPTER V

GAS-ENGINE FUELS IN LIQUID FORM — RELATION OF FUEL TO THE SIZE OF THE CYLINDER — PETROLEUM FUELS — GASOLINE — KEROSENE — ALCOHOL — DENATURED ALCOHOL

Characteristics of the Mixture After it is Drawn into the Cylinder. — The **maximum power** of a gas engine depends on the mixture or charge after it is drawn into the cylinder. To increase the power of a **steam engine,** the **cut-off** is increased until the engine is taking steam at full stroke; or the steam pressure may be increased. The matter is not so simple with the gas-engine. When the cylinder is filled with the proper explosive mixture, of maximum density, then the maximum power is available and we have no means of increasing this power. This accounts for the reluctance of the manufacturers to guarantee a large overload for their engines, and the absence of an overload when a brake test is made. If an engine does give a large overload, it shows that the rated load is smaller than it should be; that at the "rated load" the engine is really running at a lighter load than it should to give the maximum efficiency.

There are at least four conditions which influence the charge of fuel, or the mixture. These are: 1, The amount of air in the charge; 2, the temperature of the charge after it is drawn into the cylinder; 3, the pressure of the charge after it is drawn in; 4, the neutrals left in the cylinder after the exhaust stroke has been completed, which mix with the entering charge.

The amount of air in the charge should be carefully regulated by proper valves before the charge passes the inlet-valve of the engine. For gaseous fuels about 15 per cent above the theoretical air required for combustion should be used at full load. The temperature of the charge depends on the type of engine and on the load. The walls and valves of an air-cooled engine will be hotter than those of a water-cooled engine. If an engine is running at full load, the cylinder and valves will be hotter than if the engine is running at a lighter load. As the charge is drawn

in, at practically constant pressure, the hot walls and valves raise its temperature and decrease its density and its heating value proportionately. Another source of heat is the **inert gas** that is left in the cylinder. The quantity of heat given up by this inert gas depends on the amount of such gas left in the cylinder. This amount is the volume of the combustion space, or the clearance volume, and is greater on low-compression than on high-compression engines.

The **pressure reduction** of the charge, as it is drawn into the cylinder, depends on the amount of inlet-valve opening, or the amount that the governing valve is open, when the engine is governed by throttling. On large, slow-speed engines the drop in pressure should not be more than 1 pound at full load. On high-speed engines the drop exceeds this and this drop is often responsible for loss of power at high speeds, particularly in automobile engines.

In the following discussion the subscript "a" will be used to denote the condition of the charge after it has been drawn into the cylinder, and "o" the atmospheric conditions. If we assume that the charge is throttled 1 pound as it is drawn into the cylinder, its density will be made less in the ratio of its absolute pressures, or

$$d_a = \frac{p_a}{p_0} d_0 = \frac{13.7}{14.7} d_0 = 0.932\, d_0,$$

where d is the density and p the pressure.

In his book on "The Modern Gas Engine and Gas Producer" Mr. A. M. Levin has computed the probable temperatures of the charge after the completed suction stroke. He has assumed wall and valve temperatures and computed the amount of heat that would flow **into** the charge during the time the charge was in contact with the walls, valves, etc. He also computed the rise in temperature due to the neutrals that remain in the cylinder. The results of these calculations are as follows when the gas and air arrive at the inlet valve with a temperature of 62°:

	Water-cooled exhaust valve and piston.	Non-water-cooled exhaust valve and piston.
$r = 4$,	$t = 103°$ F.	$t = 120°$ F.
$r = 7$,	$t = 89°$ F.	$t = 107°$ F.

where $r = \dfrac{V_a}{V_b} = \dfrac{\text{volume cylinder and clearance volume}}{\text{clearance volume}}$,

t = temperature of charge and neutrals after mixing.

Heating Value of the Mixture. — After the mixture is drawn into the cylinder, its pressure is less and its temperature higher than without the cylinder as shown in the preceding article. The ratio of the specific volume of the mixture after it has been drawn into the cylinder to the specific volume at standard temperature and pressure is

$$\frac{V_a}{V_0} = \frac{P_0}{P_a} \times \frac{T_a}{T_0} = \frac{14.7}{13.7} \times \frac{580}{522} = 1.075 \times 1.11 = 1.195,$$

when the suction pressure is 1 pound below the atmosphere, and the temperature of the mixture in the cylinder is 120° F.

If we assume "a" to be the air required per cubic foot of gas, and "x" the proportion of actual air to theoretical air so that $(x - 1)$ will be the percentage excess air, then the volume of cold mixture per cubic foot of gas will be $(xa + 1)$ and the volume after it is drawn into the cylinder will be $\frac{V_a}{V_0}(xa + 1)$. If H is the heating value per cubic foot of gas under standard conditions, the heating value of the mixture per cubic foot after it is drawn into the cylinder will be $\frac{H}{\frac{V_a}{V_0}(xa + 1)}$.

Petroleum Fuels. — Petroleum is found in large quantities in Pennsylvania, Ohio, West Virginia, California, Texas and Russia, and to some extent in the East Indies. As it comes from the ground it is a mixture of many different hydrocarbons, with some sulphur and other impurities.

In the process of refining the light oils are driven off by distillation at comparatively low temperatures. The first vapor given off is called petroleum ether and is only a fractional per cent of the whole. This ether is all driven off below 140 degrees. Next come gasoline, the naphthas, kerosene and lubricating oils in order. After distillation there is left about 18 per cent of paraffine wax and residuum.

By using the cracking process, or destructive distillation, a considerable quantity of oils of a composition between kerosene and lubricating oil, is converted into hydrocarbons of lower density and boiling point, and thus made suitable for fuel and illuminating purposes.

The fuels that are light enough and have vapor pressures high enough to form an explosive mixture with air at atmospheric

temperatures compose about 15 per cent of the petroleum. The demand for motor fuels, caused by the increase in the number of motor cars during the past ten years, has made it necessary to resort to the cracking process and also to devise some means of using the heavier fuels in automobile motors.

TABLE V

AVERAGE ANALYSES OF AMERICAN FUEL OILS

	Gasoline.	Kerosene.	Crude oil.	Fuel oil.
Baumé degrees	70.2°	47°	18.3°	.....
Specific gravity	0.704	0.863	0.877	0.939
Per cent carbon	82.50	84.00	84.24	83.04
Per cent hydrogen	14.90	14.14	13.44	11.58
Calorific value per pound (calculated) High	21,270	20,892	20,162	19,180
Calorific value per pound (calculated) Low	19,820	19,600	19,160	18,180
Calorific value per pound (actual test) High	19,740	20,849	19,560	19,121
Calorific value per pound (actual test) Low	18,500	18,755	18,636	.....
Theoretical air, cu. ft. per lb.	193.5	189	187	178
Per cent nitrogen	1.93	1.84	0.2	0.60
Per cent oxygen	0.31	2.0	1.93	2.82
Per cent sulphur	0.09	0.23	0.823	0.90
Boiling point, degs. F.	167	437	.....	.....
Flash point, degs. F.	55	130	.....	.....

Gasoline as a Fuel.—The average gasoline found today in the markets is from 62 to 68° Baumé or 0.73 to 0.71 specific gravity. The average composition is about as follows:

Carbon,	83.5 per cent
Hydrogen,	15.5 per cent
Nitrogen, Oxygen, Sulphur, etc.	1.0 per cent
	100.0

The Smithsonian Physical Tables give the heating value of 0.71 — 0.73 specific gravity gasoline as 19,980 — 20,520. This, of course, is the higher heating value; the lower would range from 18,580 to 19,120 B.t.u. per pound. The weight of air required per pound would be

$$\text{Carbon}, \quad 0.835 \times \frac{2.67}{0.23} = 9.68$$

$$\text{Hydrogen}, \quad 0.155 \times \frac{8}{0.23} = 5.40$$

$$\text{Total} \quad 15.08 \text{ pounds.}$$

At 62 degrees and atmospheric pressure this air would have a volume of $15.08 \times 13.14 = 198$ cubic feet.

The volume of one pound of gasoline vapor is about 4.2 cubic feet at 62° F. and atmospheric pressure, and according to the analysis this requires 15 pounds or 198 cubic feet of air for theoretical combustion. If we allow some excess air, say 15 per cent, as we should with all fuels, the total air per pound of gasoline will be 17.25 pounds.

From Table IV it will appear that the saturation pressure for automobile gasoline, specific gravity 0.703, is 251 millimeters of mercury at 86° F. The pressure of the atmosphere is, on the other hand, about twice this pressure, but the gasoline vapor is about three times as heavy as air. Therefore, while a saturated mixture of gasoline vapor and air contains twice as much air as gasoline, by volume, the gasoline is 50 per cent heavier than the air. Such a mixture is too rich to explode and must be diluted with at least ten times as much air by volume as the saturated mixture contains before it becomes explosive. On account of the high vapor tension of gasoline at the higher atmospheric temperatures, there is no trouble in forming an explosive mixture, since mixtures of air and gasoline in the ratio of 17 to 1 by weight may remain saturated at temperatures below 32° F., and a very simple carburetor will therefore suffice. However, although the atmospheric temperature is high, say 86 degrees, the gasoline in the carburetor will not be vaporized at that temperature but at one much lower. The latent heat of gasoline is about 180 B.t.u. and this heat must be supplied by the 18.25 pounds of mixture resulting from the vaporization of 1 pound of gasoline with 15 per cent excess of air. Hence, each pound of mixture must supply about 10 B.t.u. of the 180 required, and this will lower the temperature to about 46 degrees, as the specific heat of the mixture will be in the neighborhood of 0.25.

If the air is at 62 degrees, the temperature of the charge after vaporization, but before it goes into the cylinder, would be about 22 degrees, if no heat is supplied from an outside source. However, carburetors are usually jacketed with hot water, when the engine is water cooled, and the air for vaporization is taken from a jacket surrounding the hot exhaust pipe. Thus the temperature of the mixture may be 60 degrees when it leaves the carburetor. As it is drawn into the engine, it will be heated up by the hot

valves and cylinder walls to a temperature of about 125 degrees. The ratio, then, of the specific volumes of the mixture after the completed suction stroke and its specific volume at standard conditions will be

$$\frac{V_a}{V_0} = \frac{P_0 T_a}{P_a T_0} = \frac{14.7}{13.2} \times \frac{585}{522} = 1.24, \qquad (1)$$

assuming that the drop in pressure will be 1.5 pounds going into the cylinder, as it might easily be with a high-speed engine.

Kerosene.—Like gasoline, kerosene is distilled from petroleum but is heavier than gasoline, has a higher flash point and lower vapor pressure of saturation for the same temperature. The specific gravity is from 0.79 to 0.82 or from 47 to 41° Baumé at 62 degrees.

The composition is about as follows:

$$\begin{aligned} C &= 0.845 \\ H_2 &= 0.155 \\ &\overline{1.000} \end{aligned}$$

The heating value is from 19,800 to 20,160 B.t.u. for the higher and 18,400 to 18,760 for the lower.

Since kerosene has a higher flash point and is less readily vaporized than gasoline, it is a safer fuel to use and to transport, but it does not make a convenient fuel for motor purposes. At ordinary temperatures a saturated mixture with air will not be explosive, so the fuel and preferably the air too must be heated to such a temperature that the process of vaporization will not bring the temperature of the mixture down below 80° F. This would result in a temperature of approximately 160° F. after the charge is drawn into the cylinder. If, as before, we assume the suction pressure to be 1.5 pounds below the atmosphere, we will have for the ratio of specific volumes

$$\frac{V_a}{V_0} = \frac{14.7}{13.2} \times \frac{620}{522} = 1.3. \qquad (1)$$

One pound of the fuel vapor occupies about 2.5 cubic feet at 62° F. and requires for combustion 190 cubic feet of air. The volume of air per cubic foot of vapor will be

	76 cu. ft.	(*a*)
Allowing 15 per cent excess	11.4 cu. ft.	
Total air per cu. ft. vapor	87.4 cu. ft.	(*xa*)

If we take 18,400 as the lower heating value, the heating value per cubic foot of vapor H will be

$$\frac{18{,}400}{2.5} = 7370 \text{ B.t.u.} \qquad (2)$$

Crude and Fuel Oil.—Crude oil is not used to any great extent for internal combustion engine fuel as it contains valuable light fuels which can be distilled off and sold for a much higher price than they would bring if sold in crude oil. **Fuel oil,** which is used in oil engines, is heavier than crude, having a specific gravity of 0.92 to 0.94, whereas the specific gravity of crude oil is in the neighborhood of 0.88. Fuel oil is a little lower in carbon and hydrogen and correspondingly higher in nitrogen, oxygen and sulphur than the crude. In other words, fuel oil might be classed as the skim milk of petroleum. However, it is fairly good skim milk as the heating value is hardly 1000 B.t.u. less than the crude oil from which the fuel is obtained. Table V shows the higher heating value of crude oil to be 20,162—calculated, while the calorimeter gave a heating value of 19,560. These figures for the fuel oil are 19,180 and 19,121 B.t.u. respectively.

Crude, or fuel, oil is used in three ways in internal combustion engines. By the first method oil is forced into a hot chamber or vaporizer which forms part of the clearance space, during the compression stroke. This vaporizer is so arranged that it is not cleared of combustion products during the exhaust stroke, but the oil is vaporized by heat in an atmosphere of inert gases. As the compression pressure builds up, pure air is forced back into the vaporizer and when air enough to make an explosive mixture has been thus forced in, an explosion occurs. Low compression, 60 to 70 pounds per square inch, is used on this type of engine, so the thermal efficiency is low.

Another method of using heavy oil is to blow it into the clearance space, by means of high-pressure air, at the end of the compression stroke, and utilize a hot tube or pot to ignite the mixture thus formed. This hot tube retains heat from the previous explosion and there is no difficulty in maintaining a temperature far above the ignition temperature of the fuel. The compression used is much higher than with a gas engine, as it ranges from 150 to 300 pounds. The fuel is burned partially at constant volume and partially at constant pressure. The maximum pressure reached with 250 pounds compression pressure may be from 400

to 600 pounds, varying with the time of fuel injection. In tests on a two-cycle engine with compression at 260 pounds gauge, the author found maximum pressures reaching up to 580 pounds, with a mean effective pressure of 137 pounds. The fuel used was Pennsylvania fuel oil. This fuel was injected into the engine cylinder with the crank 10 degrees before dead center on the compression stroke.

A third method of using heavy oil is to inject it into the cylinder with high-pressure air, at the end of compression, as in the Diesel engine. Compression is usually carried to 500 pounds and the resulting temperature is sufficient to fire the oil without the use of an ignition system. The fuel is burned at constant pressure if the injection is regulated properly. If the injection is early, due to leaky valves or faulty setting, the maximum pressures reached run as high as 1000 pounds. Due to the high compression, efficiencies are very high. Tests have been reported where the efficiency reached 52 per cent, based on indicated horse-power. A more conservative figure would be from 42 to 45 per cent.

The theoretical air required per pound of crude oil is about 190 cubic feet, while for the average fuel oil, it is in the neighborhood of 180 cubic feet. The excess air should be greater in crude-oil engines than in kerosene, gasoline or gas engines. It should be at least 50 per cent in order to get perfect combustion.

The method used in determining the size of the cylinder for a given horse-power will be given in the design of the oil engine.

Alcohol.—There are two kinds of alcohol that are available for engine fuel, ethyl, or grain alcohol (C_2H_6O), and methyl, or wood alcohol (CH_4O). Both these alcohols have a specific gravity of about 0.8 at 62° F. The boiling point of the former is 173° and the latter 148° F.

For the complete combustion of these fuels there is required:

	Oxygen per pound of fuel	Air per pound of fuel
C_2H_6O	2.086	9.03
CH_4O	1.500	6.50

The pure alcohols have heating values of about 16,000 B.t.u. per pound for the ethyl and 13,000 B.t.u. for the methyl. The manufactured alcohols never are pure but contain from 10 to 40 per cent water. With 14 per cent water the heating values would be about 13,700 and 11,000 B.t.u. respectively.

An internal revenue tax of $2.20 per gallon on 100 per cent alcohol prevented the use of this fuel for the various arts and industries for which it is suited, until this tax law was repealed about 1906. Under the new law alcohol, which has been properly denatured by the addition of a definite amount of poisonous and repulsive ingredients to render it unfit for drinking, is exempt from the payment of any internal revenue tax. By-law the "denatured alcohol" is to be made as follows: "to 100 parts of ethyl or grain alcohol by volume there must be added 10 volumes of methyl or wood alcohol and ½ volume of a heavy hydrocarbon, pyridine—denaturation benzol."

The latter ingredient is added for the purpose of giving the fuel a distinctive color and odor that can readily be recognized at inspection.

Another fuel, carbureted alcohol, is sometimes used. This is a mixture of the denatured alcohol with benzol, the latter being a purer article than that used for denaturing purposes. This benzol has a heating value a little lower than gasoline.

In France tests have been made with denatured alcohol and carbureted alcohol for motor use to determine their relative value. The analyses of the two fuels were as follows:

	Denatured alcohol.	Carbureted alcohol 50 per cent benzol.
Carbon.....	0.437	0.690
Hydrogen....	0.114	0.095
Oxygen.....	0.303	0.146
Water.....	0.146	0.069
	1.000	1.000

Computed higher heating value.....9,938 B.t.u.....14,225 B.t.u.
Heating value by test..... 10,630 B.t.u.....14,180 B.t.u.

Some of the results obtained at these tests are given below, the figures being the relative fuel consumption by weight.

Power developed, B.H.P.	Denatured alcohol.	Carbureted alcohol 50 per cent benzol.
8.3	10	7.66
16.3	10	6.85
34.4	10	7.61

During 1906 the United States Department of Agriculture conducted a series of tests with alcohol as fuel in various types of engines to determine what difficulties would be encountered in using alcohol as a fuel for the types of engines already in use; to ascertain whether the engines could be made to operate satisfactorily on alcohol fuel, and what the consumption of alcohol would be compared with the fuel for which the engine was built; and finally, to throw light upon the action of alcohol as an engine fuel, so that some information might be obtained as to the nature and extent of modifications necessary in the engine mechanisms, in order that the most economical consumption of alcohol fuel might be obtained.

The following are some of the general conclusions reached as a result of the investigations:

1. Any gasoline engine of the ordinary types can be run on alcohol fuel without any material change in the construction of the engine. The only difficulties likely to be encountered are in starting and in supplying a sufficient quantity of the fuel—a quantity which must be considerably greater than the quantity of gasoline required.

2. When an engine is run on alcohol its operation is less noisy, its maximum power is usually materially higher, and there is no danger of any injurious hammering such as may occur with gasoline.

3. Alcohol seems to be especially adopted as a fuel for automobile air-cooled engines, since the temperature of the engine may rise higher before auto-ignition takes place than is possible with gasoline fuel, and if premature ignition of alcohol does occur, no injurious hammering can result.

4. With any good small stationary engine as small a fuel consumption as 0.70 pound of gasoline or 1.16 pounds of alcohol per brake horse-power hour may reasonably be expected under favorable conditions. These values correspond to 0.118 and 0.170 gallon, respectively, or 0.95 pint of gasoline and 1.36 pints of alcohol. Based on the high calorific values of 21,120 B.t.u. per pound of gasoline and 11,880 per pound of alcohol, these consumptions represent thermal efficiencies of 17.2 per cent for the gasoline and 18.5 per cent for the alcohol.

Calculated on the basis of the low calorific value of 19,660 B.t.u. for the gasoline and 10,620 for the alcohol, the thermal efficien-

cies become 18.5 per cent for the gasoline and 20.7 for the alcohol. The ratio of consumptions is, alcohol to gasoline, 1.66 by weight, or 1.44 by volume.

Carburetion of Alcohol.—It is of special interest to determine the minimum temperature at which explosive mixtures of alcohol and air may be formed. This temperature may be determined by using Table VI, remembering that the ratio of pressures of the air and fuel vapor in a saturated mixture is equal to the ratio of the volumes of the two gases present.

TABLE VI

VAPOR PRESSURES OF ALCOHOL IN MILLIMETERS OF MERCURY

Temperature		Denatured alcohol.	Carbureted alcohol.	Temperature		Denatured alcohol.	Carbureted alcohol
° C.	° F.			° C.	° F.		
0	32	15	43	16	60.8	39.5	90
1	33.8	16	45	17	62.6	42	93.5
2	35.6	17	48	18	64.4	45	97.5
3	37.4	18	50	19	66.2	48	102
4	39.2	19	52.5	20	68.9	51	106.5
5	41	20	55	21	69.8	54	111.5
6	42.8	21.5	57.5	22	71.6	57	117.5
7	44.6	23	60.5	23	73.4	61	123.5
8	46.4	24.5	63	24	75.2	64	130
9	48.2	26	66.5	25	77	68	136.6
10	50	27.5	70	26	78.8	72	145
11	51.8	29	72.5	27	80.6	76.5	.....
12	53.6	31	75.5	28	82.4	81	.....
13	55.4	33	79	29	84.2	86	.....
14	57.2	35	82.5	30	86	92	.....
15	59	37	86	.....	.....	.....	.....

If we assume that vapors have densities theoretically due to their molecular weight, we may apply a close approximation of the Laws of Mariotte and Gay-Lussac.

If V = the volume of air in cubic feet required for the combustion of 1 pound of fuel at 32 degrees and 760 millimeters,
t = temperature of the mixture of air and vapor, above 32° F.,
x = the vapor pressure corresponding to that temperature,
d = the density of the vapor at 32 degrees and 760 millimeters,
a = coefficient of expansion, by heat of the air and vapor.

Then
$$V(1+at)\frac{760}{760-x} = \frac{1}{d}\frac{760}{x}\frac{(1+at)}{1},$$

since the volumes of air and vapor are equal in the mixture, or

$$\frac{V}{760 - x} = \frac{1}{dx}$$

and

$$x = \frac{760}{1 + dV}. \qquad (1)$$

Practice has shown that the excess air should be 50 per cent for alcohol instead of 15 per cent. Thus, denatured alcohol requires, by analysis, 7 to 8 pounds of air per pound of fuel, but experience has taught us to use 11.7 pounds. For 50 per cent carbureted alcohol the figure is 15.9 pounds, against 10.6 pounds by analysis.

Using formula (1) we find that the vapor pressure of the denatured alcohol with 50 per cent excess air is 48.6. For the 50 per cent carbureted alcohol with 50 per cent excess air, the vapor pressure is 23. Referring to Table VI we find that the lowest temperature at which the mixture can exist is 67° F. for the denatured and 12 degrees for the carbureted fuel.

If we consider the pressure of suction, say 13.2 pounds absolute, instead of atmospheric pressure, the above vapor pressures will become 43.7 and 20.7 millimeters respectively, and the temperatures of saturation 63° and 9° F.

If we require the fuel to be completely vaporized before it enters the cylinder, the heat of vaporization must be supplied by the air and fuel, which would correspond to using an unheated carbureter.

Taking denatured alcohol first we find that the latent heat at 62 degrees is 525 B.t.u. per pound. The specific heat of the gas is about 0.455 and of the air 0.24. The specific heat of the mixture will then be

$$\frac{0.24 \times 11.7 + 0.455}{12.7} = 0.257.$$

The fall in temperature of the mixture to supply the heat will then be

$$\frac{525}{0.257 \times 12.7} = 161° \text{ F.}$$

Since we must keep the mixture above 63 degrees, the initial temperature of the fuel and air will be

$$161 + 63 = 224° \text{ F.}$$

This shows the necessity of preheating the air and alcohol, and also why it is impossible to start an engine cold. If the 50 per

cent carbureted alcohol is used, calculations similar to those above show that the fuel and air must be heated only to 92° F. The specific heat of the carbureted mixture is 0.248 B.t.u. and the heat of vaporization is 350 B.t.u. The weight of mixture, per pound of fuel, is 16.9 pounds, and the temperature of saturation is 9° F.

It has been found by experiment that it is not necessary to heat the fuel and air as high as the preceding equations would appear to indicate. If the fuel is only partially vaporized in the carbureter, the vaporization may be completed by the hot walls of the engine and the hot neutrals, with marked success.

Since alcohols vaporize more slowly than gasoline, ports and valves should be large, to allow low velocities and thus have the air and fuel remain in contact for an appreciable time.

EXERCISES

1. Find the value of $\frac{V_a}{V_0}$ when the temperature in the cylinder at the end of suction is 114° F. and the pressure is 1.7 lbs. below the atmosphere.

2. Find the heating value per cubic foot of suction displacement of a gas whose heating value per cubic foot at standard condition is 137 B.t.u., air per cubic foot of gas = 1.1 cu. ft., $\frac{V_a}{V_0} = 1.35$. Allow 20 per cent excess of air.

3. Using the formula given in Chapter V for the heating value of liquid hydrocarbons, find the heating values of the kerosene and gasoline given in Table VI.

4. Find S_1 for denatured alcohol when

B.t.u. per pound = 10,450,
Theoretical air = 102.5 cu. ft. per lb.,
Volume 1 pound vapor = 9.4 cu. ft.,
Excess air = 50 per cent,
T_a = 63° F.,
P_a = 684 mm. mercury, absolute,
Compression = 186 lbs. gauge.

CHAPTER VI

GASEOUS FUELS

Natural Gas—Anthracite Producer Gas—Bituminous Producer Gas—Illuminating Gas—Coke-oven Gas—Blast-furnace Gas

General.—The gases employed as fuels in internal combustion engines are of three distinct classes. The first is natural gas; the second, manufactured gas—that is, gas that is manufactured especially for use in making power or light; and the third class includes all by-product gases.

The first class includes only one and that is the "natural" gas which is used so extensively in some parts of the world, notably in western Pennsylvania and New York, West Virginia, Ohio, Indiana and some other parts of the United States. The second includes illuminating gas, either water or bench gas, and producer gas made from anthracite or bituminous coal or oil. The third includes principally coke-oven and blast-furnace gas.

In order to give some idea of the constituents of the various gaseous fuels Table VII has been prepared and inserted here. The heating values, high and low, were calculated for the most part by using the heating values of the constituent gases as given in Table 1. The air required was calculated by using Table II.

Natural Gas.—This gas is the most convenient of any for use in internal combustion engines. It requires no cleaning, it usually contains no sulphuric or other acid-forming constituents, and is very rich in heat-producing material. The supply is being exhausted rapidly, however, and in many localities where it is used, it is not to be relied upon when the demand is very great. The gas is usually sold at a price that makes it available for power purposes.

Anthracite Producer Gas.—Anthracite producer gas, as shown by the analysis in Table VII, requires 1.12 cubic feet of air per cubic foot of gas for perfect combustion. The heating value of the gas per cubic foot at 62° F. is 136 B.t.u.

TABLE VII
Data on Common Fuel Gases

Kind of gas.	Constituents of gas in per cent volume.							Heating value of gas B.t.u. per cu. ft. at 62° F.		Cu. ft. of air required per cu. ft. of gas. Theoretical.
	H_2	CO	CH_4	C_2H_4	O_2	CO_2	N_2	High.	Low.	
Natural Gas:										
Olean, New York	.....	0.50	96.5	1.00	2.00	....	.....	998	896	9.33
Fostoria, Ohio	2.04	0.55	92.84	0.20	0.35	0.20	3.82	954	855	9.00
Pittsburgh, Pa	26.16	0.80	65.25	6.30	0.80	0.60	.....	852	763	7.50
Caspian Region, Russia	.....	.....	92.24	4.26	....	3.50	.....	1004	904	9.40
Pechelbroun, Germany	.....	3.50	77.40	4.80	1.80	3.60	8.90	874	788	8.21
Producer Gas:										
Anthracite coal	20.0	25.0	.....	.....	0.50	5.00	49.5	147	136	1.12
Bituminous coal	10.0	23.0	3.0	0.5	0.50	5.00	58.0	146	137	1.20
Coke	10.0	29.0	.....	.....	0.50	4.50	56.0	127	122	0.98
Illuminating Gas:										
Blue water gas	50.00	43.25	0.50	.....	....	3.00	3.25	312	283	2.29
Carbureted water gas	40.00	19.00	25.00	8.50	0.50	3.00	4.00	583	528	5.03
Coal or bench gas	46.00	6.00	40.00	5.00	0.50	0.50	2.00	656	585	5.80
Oil gas	32.00	.....	48.00	16.50	0.50	....	3.00	859	775	7.74
London, England	47.90	3.70	38.60	4.50	0.30	....	5.00	633	562	5.59
Coke-oven gas	53.00	6.00	35.00	2.00	....	2.00	2.00	580	514	5.06
Coke-oven gas	50.00	6.00	36.00	4.00	0.50	1.50	2.00	613	546	5.38
Blast-furnace gas	2.96	25.84	0.54	.....	....	9.37	56.00	99	97	0.76
Blast-furnace gas	5.20	26.80	1.60	.....	0.20	8.20	58.00	120	114	0.92

Bituminous Producer Gas.—The bituminous producer gas given in Table VII has a low heating value of 137 B.t.u. at 62° F. The air required is 1.2 cubic feet per cubic foot of gas, theoretical. This is more than is required for the anthracite producer gas on account of the presence of methane and ethylene, neither of which is to be found in the anthracite gas.

Illuminating Gas. This gas is used only for small engines, or for engines that run infrequently. The gas is clean, high in heating value and always ready for use but the cost is prohibi-

FIG. 19.—Pumping Station for High-pressure Fire Service.

tive. The price is seldom below 80 cents per thousand cubic feet and that means very expensive fuel. It is used to some extent in emergency high-pressure pumping stations for fire service. Usually the engines are of the multicylinder, vertical type, and are geared to triplex plunger pumps. The Coney Island, N. Y., high-pressure pumping station contains three 175 horse-power Nash vertical three-cylinder engines geared to Gould triplex pumps. The engines have cylinders 13¾ inches in diameter and a stroke of 16 inches. They make 260 revolutions per minute. The gas has a heating value of 590 B.t.u. per cubic foot, and the engines use 17.4 cubic feet of gas per brake horse-power hour when running at rated capacity. These engines received their first severe test during a fire early in 1911. The first engine was put in service 28 seconds after the signal was received, and

all three engines were in service within three minutes after the alarm was turned in. One engine ran steadily for a period of 58 hours, another ran for 23 hours without a stop and the third was run as needed. The station, during that time, pumped 9,000,000 gallons of water and consumed 186,870 cubic feet of gas.

The pumping station shown in Fig. 19 is one of two in the city of Philadelphia used for high-pressure fire service. The two stations contain 20 illuminating gas engines aggregating 5350 brake horse-power. These engines, which are geared to triplex pumps, were manufactured by the Westinghouse Machine Company.

The Nash engines at Coney Island develop

$$\frac{175}{3 \times .85} = 68.5 \text{ indicated horse-power per cylinder.}$$

The ratio of stroke to diameter is

$$\frac{16}{13.75} = 1.162.$$

The gas described as coal or bench gas in Table VII corresponds closely with the Coney Island gas. The heating value per cubic foot of suction displacement is 63.3 B.t.u.

Coke-oven Gas.—Up to about 1900 all the coke produced in the United States was made in the old "beehive" form of ovens. In these ovens the volatile matter that is distilled off is burned to keep the oven hot and to furnish heat for the coking. Since that time many "by-product" ovens have been built. As the gas is driven off from these ovens, it is led away and cooled, then washed. In these two operations all the tar is taken from the gas. The tar which is thus recovered amounts to about 40 pounds per ton of coal coked.

After the tar is removed, the ammonia is taken from the gas in one of several ways. Usually the gases are passed through weak sulphuric acid which picks up the ammonia forming the sulphate.

After the sulphate has been recovered, about half the gas is taken back to the ovens where it is burned to furnish heat for coking. The other half is available for power purposes, heating or lighting. The gas available for these uses contains about 7 per cent of the heat of the coal. Thus, the heat contained in the by-product gas per short ton of coal would be

$$2000 \times 14{,}000 \times 0.07 = 2{,}560{,}000 \text{ B.t.u.}$$

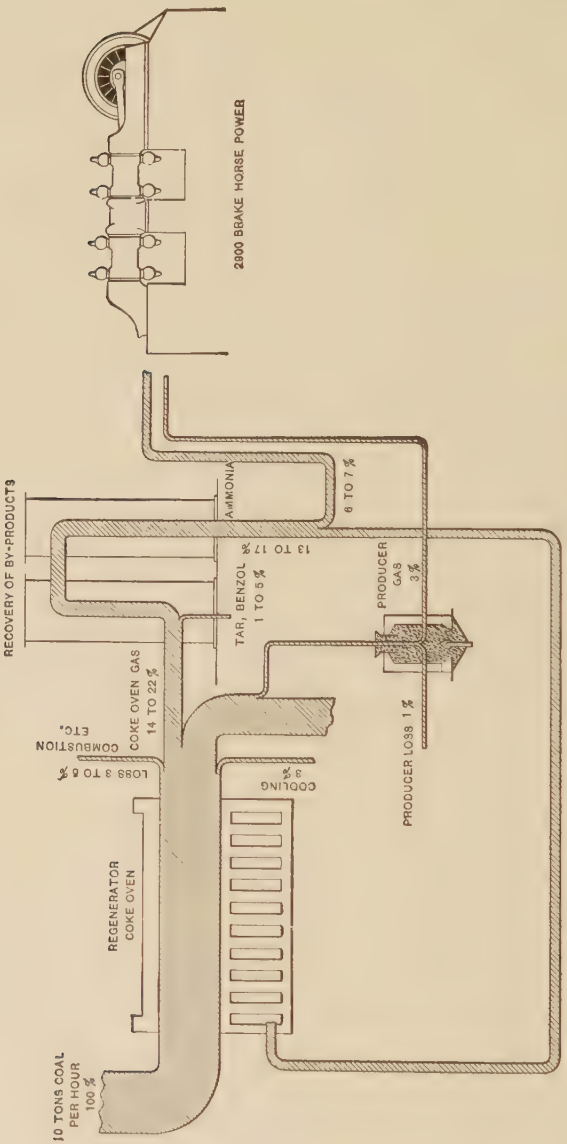


FIG. 20.—Heat Balance Diagram of a By-product Coke-oven.

If the efficiency of a gas engine be taken as 25 per cent, the power to be derived from this gas would be

$$\frac{2{,}560{,}000 \times 0.25}{2545} = 251 \text{ horse-power hours.}$$

If this gas has a heating value of 550 B.t.u. per cubic foot the volume available for power per ton of coal coked would be 4650 cubic feet. Fig. 20 shows the heat balance of a battery of coke ovens of a capacity of 500 tons per day of 24 hours. The by-product gas, if used in gas engines, would generate 2000 brake horse-power continuously. If the coke "smalls" (refuse coke) were used in a gas-producer, and this gas mixed with the coke-oven gas, an additional 900 horse-power could be generated.

Blast-furnace Gas.—The first attempt to use blast-furnace gas in gas engines on a large scale in the United States was in 1903 at the Lackawanna Steel Company, Buffalo, N. Y. This plant consisted of sixteen 2000 horse-power blowing engines and eight 1000 horse-power generating engines. All the engines are of the twin, two-cylinder, two-cycle horizontal type, built by the De La Vergne Machine Company of New York working on the Koerting patents.

One of the houses containing eight of the 2000 horse-power blowing engines is shown in Fig. 21. While the gas engines are horizontal, the blowing cylinders, of which there is one for each engine, are vertical, placed on *A* frames over the right-hand main shaft bearing. The blowing-cylinder connecting-rod is on the same crank-pin with the right-hand motor-cylinder rod.

These engines did not give entire satisfaction when first started, but they were subject to much adverse criticism that was entirely unjust. As is the case in any new departure in the way of large power installations, it took some time to get these engines running satisfactorily, but the fact that they are giving better service in their eleventh year than they did in their first year of life is conclusive evidence that they are not failures by any means.

The largest installation of blast-furnace gas engines in this country up to the present time is that of the Indiana Steel Company, Gary, Indiana. In this plant there have been installed 16 blowing-engines of the twin-tandem, four-cylinder, horizontal, four-cycle type. These engines have four power-cylinders each, 42 by 54 inches, and two blowing-cylinders, the capacity of the latter being 30,000 cubic feet of free air per minute. Eight of

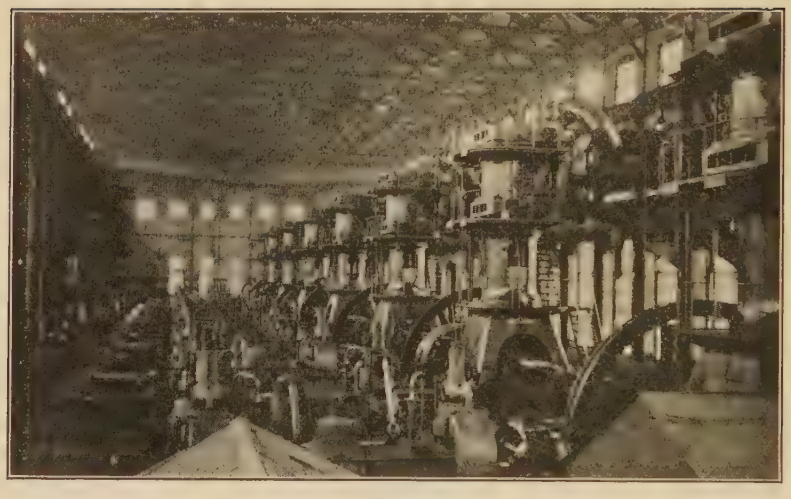


FIG. 21.—De La Vergne-Koerting Gas Blowing Engines. Lackawanna Steel Co., Buffalo, N. Y.

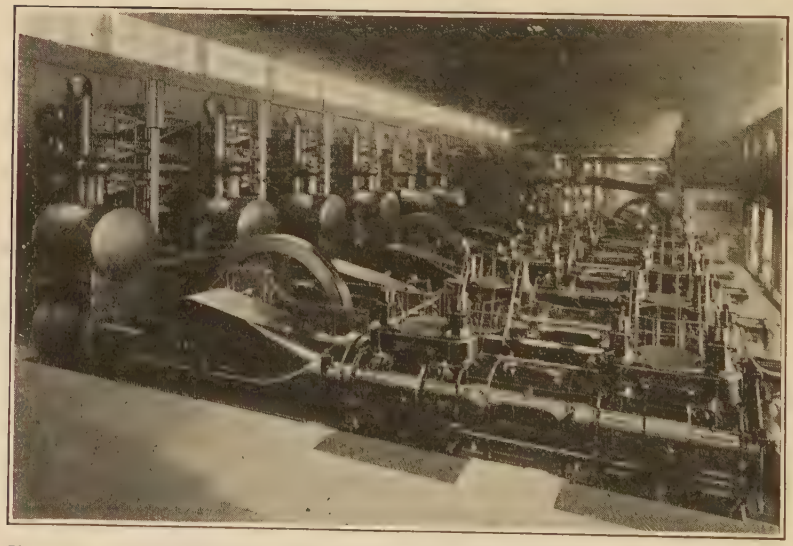


FIG. 22.—Allis-Chalmers Gas Blowing Engines. South Chicago Works, Illinois Steel Co.

these engines were built by the Allis-Chalmers Company and eight by the Westinghouse Machine Company.

In the central power station of the Gary plant there were originally installed 17 Allis-Chalmers engines of the same size as the blowing engines above. Fifteen of these engines are direct connected to 25-cycle, three-phase, 6600-volt generators and two are connected to 250-volt direct-current generators. The engines are rated at 4000 horse-power at $83\frac{1}{3}$ revolutions per minute, the generator at 2000 kilowatts. These are shown in Fig. 23.

Fig. 22 shows three 48 and 78 by 60 twin tandem gas blowing engines built by Allis-Chalmers Company for the Illinois Steel Co., South Chicago, Ill.

The standard blast-furnace, in the United States, has a capacity of 500 tons of iron every twenty-four hours. The heat balance of such a furnace is shown diagrammatically in Fig. 24. Of the 604 million heat units charged per hour in the form of coke 307 million are absorbed in the furnace proper, and 89 million in the stoves that heat the blast. Deducting 15 million for auxiliaries and washers 193 million are left for gas engines. From this heat 19,200 brake horse-power could be generated and of that amount 3600 would be required for furnishing the blast; the balance, 15,600, would be available for general power purposes.

The most expensive part of the blast-furnace gas power installation, aside from the engines themselves, is the cleaning system. This system is elaborate, cleaning the gas by both dry and wet scrubbing. The Gary plant is up-to-date in this respect and a short description of that cleaning plant will not be out of place here.

Cleaning Blast-furnace Gas.—The gas is discharged from the top of the furnace through four outlets into two large pipes known as the "downcomers," which lead to a reservoir 30 feet in diameter by 40 feet high, called the "dry dust catcher." There a considerable part of the impurities settles to the bottom and the gas passes into another large pipe emptying into a supplemental tank 14 feet in diameter by 25 feet high, one of this size serving each pair of furnaces. This structure not only provides an additional dust catcher but acts also as a valve, being divided into two compartments partially filled with water. By increasing the height of the water in either one, the furnace on that side can be cut off as desired, and there will be no back-flow from the mains

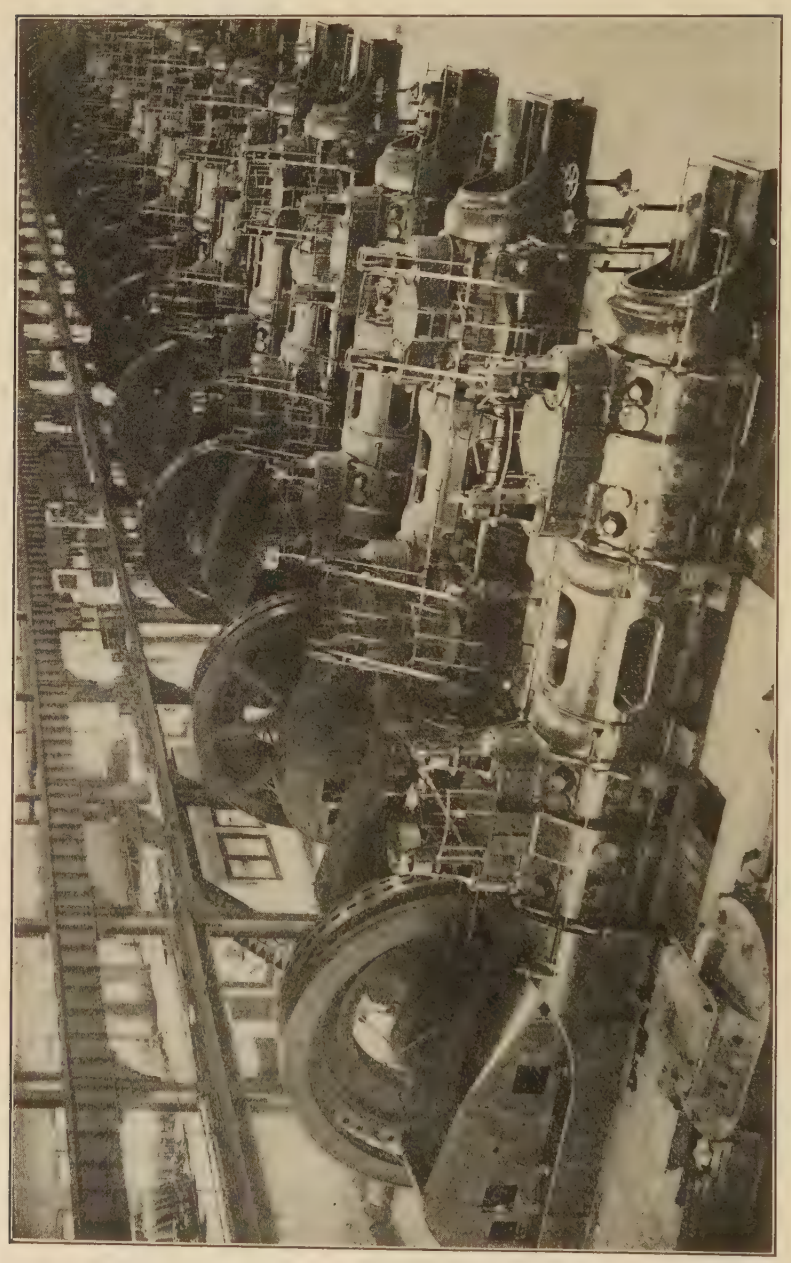


FIG. 23.—Allis-Chalmers Gas Engines Driving Electric Generators. Indiana Steel Co., Gary, Ind.

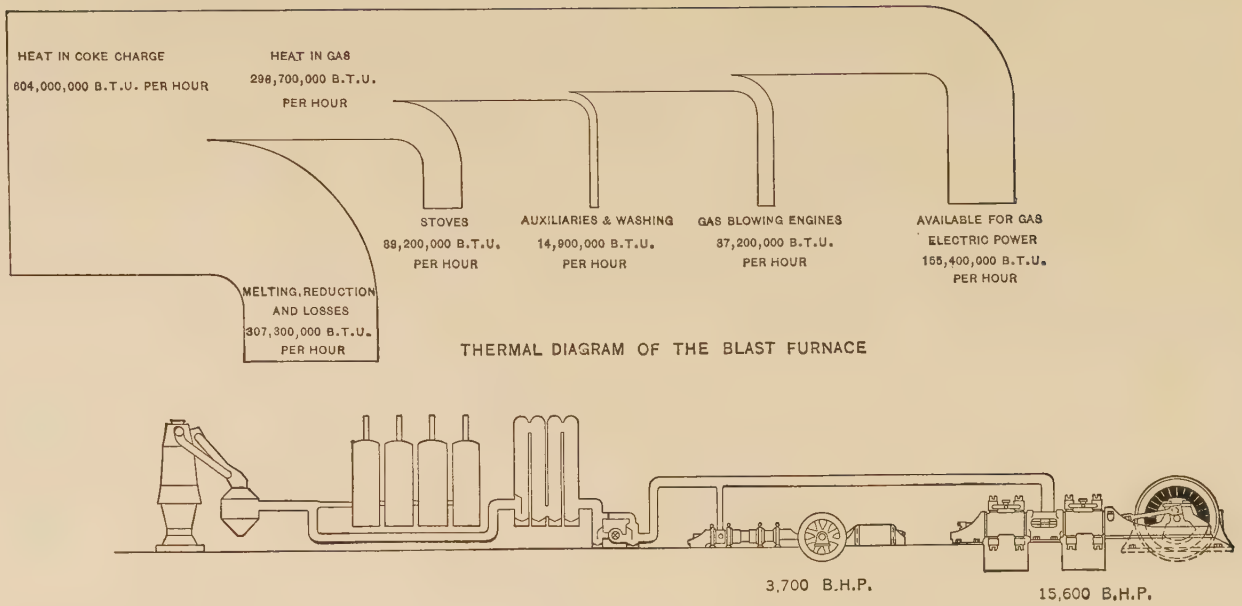


FIG. 24.—Heat Balance Diagram of a Blast Furnace.

beyond. The two chambers of this tank discharge the gas into a ten-foot pipe, which carries the gas and the remaining impurities into the primary wet washers. There are three of these to each pair of furnaces, and each one has the capacity to take care of the gas from a single furnace, thus providing a spare washer for cleaning or repairing. These primary washers are cylindrical in form with conical top and bottom, and are about one-third full of water, an over-flow maintaining the proper level. Here the gas is discharged against the surface of the water from pipes with fluted ends, and then escapes around these ends into the openings of a larger main.

From the primary washers the gas is taken to the secondary washers, the first of this group being vertical scrubbers—drums 14 feet in diameter by 50 feet high. Water is sprayed from the tops of these towers, and the gas is let in near the bottom and rises up through the falling water. At the top the gas passes out and goes to the Thiessen washers, four of which are installed for each pair of blast furnaces. In these washers the gas is led between the wall of a cylinder and a revolving drum, the drum carrying a series of paddles or blades. A stream of water is spread into a film on the surface of the cylinder and the whirling drum throws such impurities as the gas still holds against the water film where they are caught and held. From these final washers the gas is conveyed under slight pressure to the two holders, each of which has a capacity of 200,000 cubic feet. From these holders the gas is drawn, as required, by the electric power station and the blowing-engine houses.

Blast-furnace Gas as a Fuel.—Blast-furnace gas is low in heating value, varying from 95 to 105 B.t.u. per cubic foot. On very damp days in the summer, the moisture of the air is dissociated in the furnace and the hydrogen thus formed raises the heating value of the gas appreciably. The gas described in line 16, Table VII, has a heating value of 97 B.t.u. at 62° F. It requires 0.76 cubic foot of air for theoretical combustion.

EXERCISES

1. Check the higher heating values of the gases in Table VII.
2. Check the lower heating values of the gases in Table VIII.
3. Check the cubic feet of air required for the gases in Table VIII.

CHAPTER VII

THE OTTO AND DIESEL CYCLES IN PRACTICE

Comparison of the Practical Points of These Two Principal Engine Cycles

Combustion. — The steps which take place during the process of adding heat to the charge in a gas engine cylinder are first, ignition; second, inflammation; third, explosion; fourth, combustion. **Ignition** takes place when sufficient heat is communicated to a mixture of gas and air by a flame, electric spark or hot tube to raise a particle of gas to the ignition temperature. Only a small portion of the charge need be heated in this way. **Inflammation** is the subsequent spreading of the flame throughout the mixture, or its propagation from one particle to another, until the whole volume is alight. **Explosion** follows when the mixture is completely inflamed and the maximum pressure attained. When all the gas in the cylinder is thoroughly lighted, the particles are driven widely apart, and thus the moment of complete inflammation will be that of maximum pressure. Complete inflammation and explosion are thus practically simultaneous. **Combustion** is complete when all the chemical changes have taken place and the gases have been transformed into water vapor and carbon dioxide. This moment may not, and usually does not, coincide with the point of explosion in the actual gas engine cylinder. The complete combustion is almost always delayed and it is this delay which gives the expansion line a form other than the true adiabatic for air or for the gas under question. This delay is appreciable and may be measured by methods which may be used without difficulty in the laboratory.

Flame Propagation. — The rate of flame propagation in a perfectly vaporized explosive mixture of correct proportions is very high, and ordinarily there is no advantage in increasing this velocity. In diluted mixtures, however, the rate of propagation is much lower and in many cases so low as to have a deleterious effect on the efficiency of the engine. Usually this low rate of propagation is responsible for the low efficiency of gas engines at

light loads, when equipped with certain kinds of governors. Quite frequently special arrangements are made to increase the rate under these conditions.

Experiments by Mr. Grover at Yorkshire College show that if the excess air is replaced in part by burned gases, the rapidity of combustion may be increased. These experiments were made with coal gas. Experimenters are not united, however, as to the effect on the rapidity of combustion of replacing the excess air by neutrals in explosive mixtures in general. With some gases it seems to be a detriment to cut down the excess air by introducing neutrals.

Mr. Grover's experiments establish the following additional conclusions with respect to a coal gas mixture:

The highest pressures are obtained when the volume of air is slightly in excess of the amount required for complete combustion.

When the volume of the products of combustion does not exceed 58 per cent of the total mixture, the mixture is still explosive, provided the volume of air is not less than $5\frac{1}{2}$ times the volume of the gas.

Higher pressures are recorded when the neutral gases take the place of excess air.

The above conclusions, with regard to the experiments, amount practically to saying that a perfect coal gas and air mixture may be diluted with burned gas amounting to 140 per cent of its volume and still be explosive; and the velocity of the flame propagation of such a mixture is more rapid than if air were used for a diluent.

The temperature of the charge when ignition takes place also has an influence on the rate of flame propagation, so that the higher the temperature, the more rapid the combustion becomes.

Experience also shows that mixtures in which the fuel is imperfectly vaporized often give unsatisfactory results due to slow combustion, and that a charge too weak to explode with a normal spark can be made to explode with a heavy spark.

Clerk's Experiments. — Mr. Dugald Clerk, in his experiments, considered that, to understand the action of gas in a cylinder, it was necessary to determine not only the time required to obtain a maximum pressure but also the duration of that pressure.

Mr. Clerk's experiments were with a small cylinder without

piston, filled with different explosive mixtures, connected to an indicator, the drum and paper of which were made to revolve so that each tenth of a revolution occupied 0.033 second. The pressure of the explosive gases forced up the indicator pencil, and by dividing the area of the moving drum on which the pencil traced curves into sections, the time occupied by the explosion and the cooling or reduction of pressure of the gases could be estimated within $\frac{1}{100}$ of a second. On this diagram the ordinates represented pressures, and the abscissæ the time of the explosion in fractions of a second. The maximum explosion pressure was developed in a closed vessel and therefore at constant volume, and the cylinder having no piston, no heat was expended in doing work. The diagrams showed that the pressure of the gases fell more slowly than it rose. The maximum pressure was produced in 0.026 second after ignition; the fall to atmospheric pressure and temperature occupied 1.5 seconds, or nearly sixty times as long. Without previous compression of the gases, the highest pressure obtainable with a dilution of 5 parts air to 1 part gas was only 96 pounds per square inch; with compression and a much weaker mixture this pressure was nearly doubled. The critical mixture, or the weakest dilution of gas and air that will ignite, varied according to the quality of the gas used. With Oldham gas a charge of 15 parts of air to 1 part of gas ignited, and the pressure was 40 pounds per square inch above the atmosphere. With Glasgow gas the critical mixture was 14 parts of air to 1 part of gas, and the pressure produced was 52 pounds per square inch.

The results of these experiments are shown in Table VIII.

TABLE VIII

Ratio by volume.		Maximum observed pressure. Lbs. per sq. in.	Time to reach maximum pressure. Seconds.
Gas.	Air.		
1	14	40.0	0.45
1	13	51.5	0.31
1	12	60.0	0.24
1	11	61.0	0.17
1	9	78.0	0.08
1	7	87.0	0.06
1	6	90.0	0.04
1	5	91.0	0.055
1	4	80.0	0.16

Massachusetts Institute Experiments. — Trials were carried out in 1898 at the Engineering Laboratory of the Massachusetts Institute of Technology to determine the interval of time in tenths of a second elapsing between ignition and the attainment of maximum pressure, and also the pressure of explosive mixtures of lighting gas and air in different proportions at each twelfth of a second. The pressures were shown on indicator diagrams, while a marker attached to a tuning fork traced simultaneously a time wave line of vibrations equal to one-sixtieth of a second. The explosions took place in a cast-iron cylinder, without a piston, of 310 cubic inches volume, to which a mercurial gauge for recording pressures, an air and a vacuum pump, and two electric batteries for firing the charge and keeping the tuning fork in vibration were connected. The pressure-recording apparatus consisted of an indicator paper fixed on a circular revolving disc driven at a uniform speed, on which by a special arrangement the tuning fork traced a succession of contiguous lines. Thus the time and pressure were recorded simultaneously, the atmospheric line and the diagrams of pressure both being circular.

TABLE IX

RESULTS OF TESTS ON EXPLOSIVE MIXTURES OF ILLUMINATING GAS AND AIR

Mixture parts by volume, gas, air.	Maximum pressure, lbs. per sq. in.	Time of explosion, seconds.	Area, sq. in.	First one-fifth second.		Final pressure.
				Mean pressure, lbs. per sq. in.	Mean pressure ÷ proportion of gas.	
1–3	45	0.49	0.32	11	44	26
1–4	86	0.08	1.77	59	295	61
1–5	96	0.05	1.86	62	372	52
1–6	88	0.05	1.80	60	420	54
1–7	86	0.06	1.97	66	528	58
1–8	87	0.06	1.71	57	513	53
1–9	77	0.08	1.60	53	530	57
1–10	71	0.11	1.36	45	495	56
1–11	68	0.14	1.21	40	480	60
1–12	39	0.33	0.35	12	156	29
1–13	32	0.42	0.18	6	84	16
1–14	9	0.42	0.05	2	30	4

To determine the pressures the cylinder was first thoroughly cleansed, then connected to the vacuum pump, and the air exhausted to about one-sixth of an atmosphere. Into the partial

vacuum thus formed, giving a pressure of 5 inches of mercury, gas was admitted and the pressure was raised to atmospheric, thus producing a mixture of one part of gas to five of air. The charge was then fired electrically, the tuning fork set in vibration and the diagrams of pressure were taken. The proportion of gas and air could be varied by the vacuum in the pump.

The results of these tests are shown in Table IX and graphically in Fig. 25.

It will be noticed by the diagrams in Fig. 25 that, generally speaking, the more rapid the combustion and the higher the

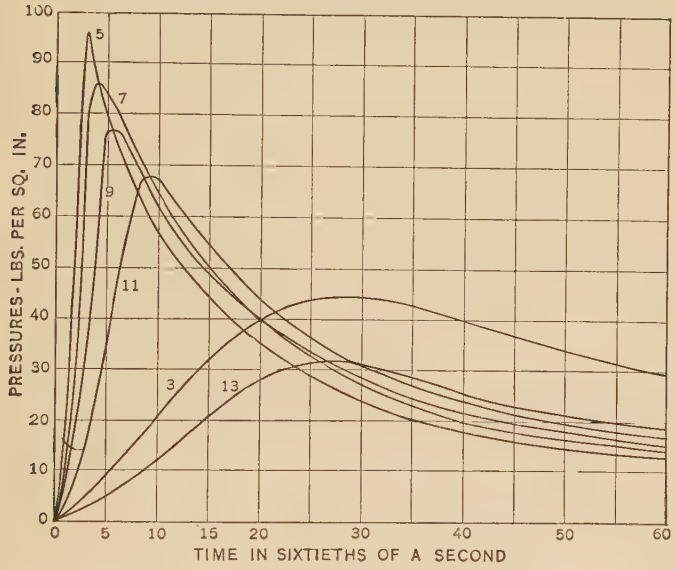


Fig. 25.

pressure, the lower the pressure line becomes after an interval of 1 second, except in the case of the very rich mixture, curve 3. The end of the first one-fifth second from the time of ignition is marked in the figure by a heavy line at the point $\frac{12}{60}$ of the total length of the diagrams, and at that line the final pressures of the first one-fifth second are measured. The areas obtained for the first one-fifth second and the mean pressures, columns 4 and 5 of Table IX, refer also to the areas of the diagrams up to that line. The figures of the sixth column represent the efficiencies of combustion for each of the mixtures.

Mixtures Diluted with Combustion Products. — It might appear from Mr. Grover's experiments with mixtures diluted with air and burned gases that the neutrals in a weak mixture would tend to facilitate rather than retard the rate of combustion. However, the conditions under which these experiments were carried out differ materially from the conditions found in actual gas engine cylinders, and experience, with some fuels at least, show results differing from those found by Mr. Grover.

Assume a case of a throttling engine with a fairly high compression, which, due to faulty valve setting, does not relieve itself of burned gases readily. It is found that such an engine will give evidence of pre-ignitions on heavy loads, which are due probably to the excess heat which the neutrals give to the fresh charge. Under a light load, however, the operation of the engine is liable to be troubled with back firing into the mixing chamber or inlet valve casing, at the time of opening of the inlet valve for the admission of a fresh charge. This feature might be explained on the ground that an excessive amount of combustion products in a weak mixture makes it slow burning, not only during the expansion stroke but also during the exhaust stroke and up to the time of opening of the inlet valve for the following suction stroke.

After readjusting the valve setting, by opening the exhaust valve early and keeping it open as long as practical, an appreciable change will have been effected. The cylinder will become scavenged from combustion products as thoroughly as possible; hence the weak mixture on the same light load as before will be diluted by more air instead of by an excess of combustion products, and the back firing into the mixing chamber will have been cured. This tends to show that of the two mixtures of similar heating value, the latter, containing less combustion products, acts, in the working cylinder, as if it were the more inflammable.

Timing the Ignition. — Cards with normal, early and late ignitions are shown in Fig. 26. Card *A* shows what might be called normal ignition, card *B* early and card *C*, late ignition. Card *A* is ideal, that is, it is a little too perfect for an actual card. Of course the shape of the card will vary somewhat with the quality of the fuel. With a lean gas, such as producer or blast-furnace gas, the actual firing of the spark will occur at some point considerably before the end of the compression stroke, to give a card similar to the one shown at *A*. This may readily be understood

if the results of the experiments on flame propagation are studied carefully. Those tests show that the maximum pressure is attained an appreciable time after the spark is fired. This accounts for the necessity of firing the spark early when a lean mixture is being used.

Card *B* shows a spark that would probably be too early except for a high-speed engine. Later on in this book will be shown the effect of an early spark at different speeds, in connection with the inertia of the reciprocating parts. In general it might be said that early ignition may, and should, be used with lean mixtures and high speeds. However, using an early ignition with lean mixtures should result in card *A* and not card *B* if the speed

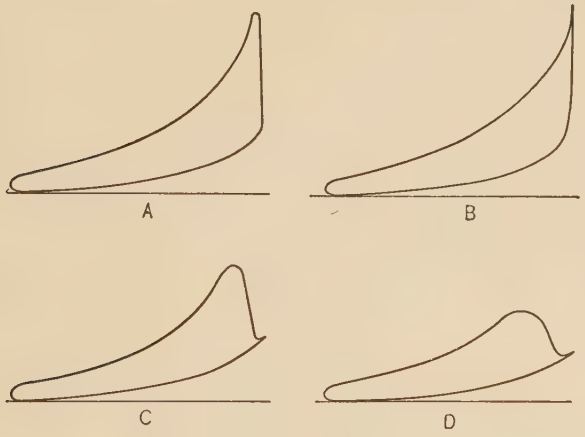


FIG. 26.

is not high. A card as shown at *B* is permissible only at high speed in order to get the cushioning effect.

Card *C* shows a diagram taken from an engine with late combustion. The cause might be any one of several, viz: — first, a late spark with normal mixture; second, a normal spark with a mixture that is too rich, as shown by curve 3, Fig. 25; third, a normal spark with a lean mixture, curve 13, Fig. 25; fourth, a cold engine; that is, when an engine is first started, even if the spark be set for normal operation, combustion would be late enough to give the card as shown at *C*.

This card *C* is indicative of low efficiency and should not be used where it can be avoided. The writer has found that in

certain engines which use a hot bulb for ignition, this card often results if too much water is injected into the cylinder. If the supply of water, or steam, where steam is used instead of water, is cut down to a minimum without producing premature explosions, much better efficiency will result and a larger load can be carried on the engine.

Suppression of Heat at Combustion. — The specific heats of gases have been found to be constant for a range of temperature up to about 400° F. Regnault has found that the specific heat at constant pressure c_p for air is 0.2375 from 0° to 200° C. Generally, both c_p and c_v for gases may be assumed to be constant up to 200° C. If we assume c_v constant throughout the range of temperature in a gas engine, then the maximum temperature of combustion will be

$$T_c = T_b + \frac{Q}{c_v},$$

where T_c is the temperature of combustion and T_b is the temperature at the end of compression, or the beginning of combustion.

The formula would give much higher temperatures than are reached in an actual engine for several reasons. The specific heat of gas is not constant but, as a general rule, the law of variation is expressed by a linear equation; thus,

$$c_v = a + bt,$$
$$c_p = a' + bt.$$

Further, some of the heat generated by the combustion is dissipated into the cylinder walls. There has been a good deal of controversy as to the relative amount of the heat that is lost in this way, but indications seem to point to only a small loss. Probably an appreciable amount is lost through dissociation of the gases at temperatures reached in actual practice, and it is certain that the generation of heat is less than the theoretical owing to the fact that combustion does not take place at constant volume. The piston moves an appreciable amount during combustion and this movement increases the farther the mixture is from the theoretical explosive mixture. That is, with a mixture that is too rich or too lean, combustion takes place during the movement of the piston when the gas is doing work. The heat equivalent of this work, then, does not go to help raise the temperature, consequently the temperature reached is lower than the theoretical.

The values for specific heats of products of combustion, at constant pressure and constant volume, are shown in Table X, the temperature t being expressed in degrees Fahrenheit. These values are those determined by Mallard and Le Chatelier.

TABLE X

SPECIFIC HEAT AT CONSTANT PRESSURE

$$CO_2 = 0.185 + 0.000093\,t,$$
$$H_2O = 0.415 + 0.000202\,t,$$
$$N_2 = 0.240 + 0.000024\,t,$$
$$O_2 = 0.211 + 0.000021\,t,$$
$$\text{Air} = 0.233 + 0.000023\,t.$$

SPECIFIC HEAT AT CONSTANT VOLUME

$$CO_2 = 0.140 + 0.000093\,t,$$
$$H_2O = 0.360 + 0.000202\,t,$$
$$N_2 = 0.173 + 0.000024\,t,$$
$$O_2 = 0.150 + 0.000021\,t,$$
$$\text{Air} = 0.165 + 0.000023\,t.$$

To illustrate the method of finding the probable combustion pressure of a fuel in a gas engine, the following example is worked out.

Let us assume a fuel, the composition of which is, by weight.

	Per cent
H_2 =	1.01
CO =	25.78
CO_2 =	17.39
CH_4 =	0.92
N_2 =	54.90
	100.00

The weight of air to burn a pound of gas will then be, from Table II,

$$H_2 = 0.01 \times 34.640 = 0.3464$$
$$CO = 0.258 \times 2.477 = 0.6390$$
$$CH_4 = 0.009 \times 17.36 = 0.1559$$
$$1.1413$$

The excess air will be

$$0.15 \times 1.1413 = 0.1712 \text{ pounds.}$$
$$0.1712 \times 0.23 = 0.0398 \text{ pounds excess oxygen.}$$
$$0.1712 - 0.0398 = 0.1314 \text{ pounds excess nitrogen.}$$

The weights of the combustion products will then be, from Table II:

	CO_2	H_2O	N	O
H_2	.....	0.09	0.2664	.....
CH_4	0.024575	0.02025	0.1199	.....
CO	0.40560	.....	0.4915	.....
CO_2	0.17400	.....	.....	.....
N_2	.....	.....	0.5490	.....
Excess	.....	.....	0.1314	0.0398
Totals per pound gas...	0.60435	0.11025	1.5582	0.0398

The maximum temperature of combustion is probably in the neighborhood of 3000° F., as an average.

The absolute temperature after compression is, if we assume a compression ratio of 6, corresponding to 135 pounds per square inch compression,

$$T_b = r^{n-1} \times T_a = 6^{0.35}\,(460 + 80) = 1.87 \times 540 = 1010 \text{ degrees}$$

or 550° above Fahrenheit zero.

The mean temperature during combustion will be about 1800 degrees. The mean specific heat of the gases in the products of combustion may then be found from Table X.

These are

$$CO_2 = 0.317,$$
$$H_2O = 0.690,$$
$$N_2 = 0.218,$$
$$O_2 = 0.190.$$

The mean c_v for the products of combustion will then be found as follows:

$$CO_2 = 0.60435 \times 0.317 = 0.1916$$
$$H_2O = 0.11025 \times 0.690 = 0.0761$$
$$N_2 = 1.55820 \times 0.218 = 0.3397$$
$$O_2 = 0.0398 \times 0.190 = 0.0076$$
$$2.3126 \qquad\qquad 0.6150$$

The C_v per pound will then be $\frac{0.6150}{2.3126} = 0.2659$.

In order to find the heating value per pound of mixture, including excess air, we may use the constituents given per pound of fuel. These must be reduced to percentages, and each fractional part in the mixture multiplied by its heating value.

	Gas and air, per lb. fuel.	Per cent weight of mixture.		B.t.u. per lb. lower value.
H_2	0.010	0.00432	×	52,275 = 258
CO	0.258	0.11150	×	4,380 = 488
CO_2	0.174	0.07519		
CH_4	0.009	0.00389	×	21,544 = 84
N_2	0.549	0.23725		
Air	1.314	0.56786		
	2.314	1.00001		830

Thus, 830 is the heating value of the fresh charge. But the fresh charge is mixed with the neutrals before combustion takes place, hence the heating value will be lowered. The heating value of the charge after being mixed with the neutrals will be

$$830 \times \frac{r-1}{r} = 830 \times \frac{5}{6} = 691 \text{ B.t.u. per pound.}$$

Under the discussion of the Otto cycle, Chapter III, the ratio of temperatures before and after combustion is given by the formula

$$\frac{T_c}{T_b} = \frac{Q}{c_v T_b} + 1.$$

Since the combustion occurs at constant volume, we may write

$$\frac{P_c}{P_b} = \frac{Q}{c_v T_b} + 1,$$

P_c being the pressure after and P_b before combustion. It has been shown that not all the heat of combustion is effective in increasing the pressure but some of it is lost in various ways. If we assume the loss to be 0.15 per cent, the available heat will be 0.85 Q. Then

$$\frac{P_c}{P_b} = \frac{0.85 \times 691}{0.2659 \times 1010} + 1 = 2.18 + 1 = 3.18,$$

$$P_c = 3.18 \times 150 = 477 \text{ pounds absolute.}$$

This result is probably not far from the pressure that would actually be attained in the engine cylinder with the kind of gas used in this example. The value of this formula, however, depends upon how accurately the percentage heat lost during combustion can be estimated. There is no reliable data or information at present on the subject, and it seems to the author that, until such data is available, this method should be used with caution. An-

other method which has been found to give satisfactory results will be discussed under the design of the engine.

The Diesel Cycle. — Dr. Diesel's original intention was to build an engine working on practically the constant temperature cycle, and though the Diesel engines work in a manner somewhat different from that which the inventor intended, it might be interesting and of value to know what the inventor tried to do in his early experiments. In the patent specification dated August 27, 1892, Dr. Diesel proposed to compress air alone to the highest temperature of the cycle,

"that is to say the temperature at which the subsequent combustion has to take place, namely, the combustion temperature (not the burning or ignition point)."

"If it be desired, for instance, that the late combustion shall take place at a temperature of 700° C., the pressure will be 64 atmospheres; for 800° C., the pressure will be 90 atmospheres, and so on."

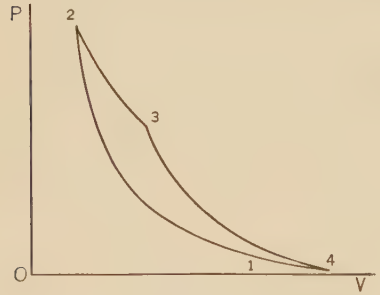


FIG. 27.

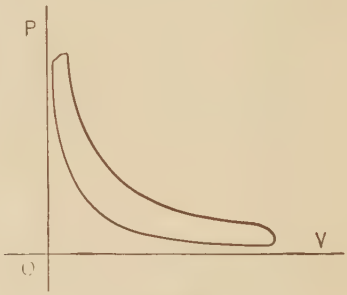


FIG. 28.

In this compressed body of air the fuel was to be injected and ignited, far above the temperature necessary for inflammation. Simultaneously with the gradual introduction of fuel, an expansion of the body of air was to take place, which was to be regulated in such a manner that the cooling caused by expansion would destroy at each moment the heat produced by the combustion of the several introduced particles of fuel; owing to this the combustion would show its effect, not by an increase of temperature but solely by work done. The diagram corresponding to this explanation is shown in Fig. 27.

Diesel found it impossible to carry out this cycle in practice owing to the high pressures involved, so he compromised on the card shown in Fig. 28. In this card, compression is carried to a

pressure that will result in a temperature high enough to ignite the fuel. This compression is in the neighborhood of 500 pounds, resulting in a temperature of 1000° F. in actual practice. The fuel is injected at a point and at a rate which will give constant-pressure combustion but not constant temperature, in fact the temperature rises until it is little, if any, lower than in the constant-volume engine. This temperature rise may continue for a short period after cut-off. This type of engine was described by Diesel in his second patent.

It is very important to note from a practical point of view that the duration of combustion is considerable and hence there is a longer time available for getting rid of the heat by the jacket water than in a gas engine, where combustion is rapid and the temperature rises very rapidly. It follows, therefore, that for the same maximum pressures in the cylinder of a gas engine and a Diesel engine the temperature rise in the first case would be higher than in the latter. Since a Diesel engine may work with much higher compression pressure than a gas engine and yet not be subjected to higher temperatures, hence the fact already well established, that greater powers may be obtained per cylinder than with the ordinary types of internal combustion engines. There are, however, other factors entering into this matter.

EXERCISES

1. Find the c_p of the following gas at 2200° F.: CO_2 = 25 per cent, H_2O = 14 per cent, N_2 = 27 per cent, Air = Bal. by weight.

2. Find the c_v of the following gas at 1500° F.: CO_2 = 23 per cent, N_2 = 52 per cent, O_2 = 10 per cent, Air = Bal. by volume.

3. Given: H_2 = 2 per cent, CO = 25 per cent, CH_4 = 22 per cent, CO_2 = 14 per cent, N_2 = 43 per cent, by weight. Find the weights of the combustion products when the gas is burned with 15 per cent excess of air.

4. Find the c_v of the combustion products of the gas given in Prob. 3.

5. Find the B.t.u. per pound of the mixture of the gas given in Prob. 3 with air for combustion and 20 per cent excess.

6. Find the probable explosion pressure of gas in Prob. 3 when r = 7. Assume efficiency of combustion to be 85 per cent.

CHAPTER VIII

IGNITION — TIMING — HOT-TUBE — MAKE-AND-BREAK — JUMP-SPARK — PARTS OF IGNITION SYSTEM

Timing the Ignition. — The ignition of any gas engine should be properly timed. The time of ignition varies with conditions, and cannot be set by any fixed rule. If electric ignition is used, the device is so arranged that the point of ignition can be varied at will within certain limits. In general the conditions that affect the ignition are: first, speed of engine; second, kind of fuel; third, quality of mixture.

The higher the speed of the engine, the earlier the spark should be set. This is easily explained by the fact that it takes an appreciable time for combustion to occur after the spark is fired. At high speeds, then, the spark must be set early in order to have the maximum pressure occur at the beginning of the stroke. Manufacturers of automobiles sometimes make arrangements for the spark to be advanced or retarded automatically as the engine speeds up or slows down. This is an advantage as it relieves the driver of the care of the timing and it also gives more accurate results than could be obtained by hand manipulation. As an automobile engine slows down, due to the increase in load, the period of time in which combustion may take place becomes lengthened, and if the spark is set early for high speed, it occurs too soon for the reduced speed, and maximum pressure is set up before the piston reaches the dead center. This trouble makes itself known, first, by knocking in the cylinder, and finally by sudden stopping of the engine due to the early high pressure.

Fuels that naturally burn **rapidly** require a later spark than the slower-burning fuels. Thus, gasoline or natural-gas engines should have later ignition than producer or blast-furnace gas-engines, if the compression is the same. But, since higher compression increases the rate of combustion, the higher the compression the later may be the ignition for a given fuel.

When an engine is running at a **light load** (with a lean mix-

ture), combustion will be slower and ignition should be correspondingly earlier. Likewise, a mixture that is too rich would require an early spark for the same reason. The theoretically proper mixtures have the most rapid combustion and require the latest spark.

Several systems of ignition are used at the present time, although the major part of all engines being made today have an electrical

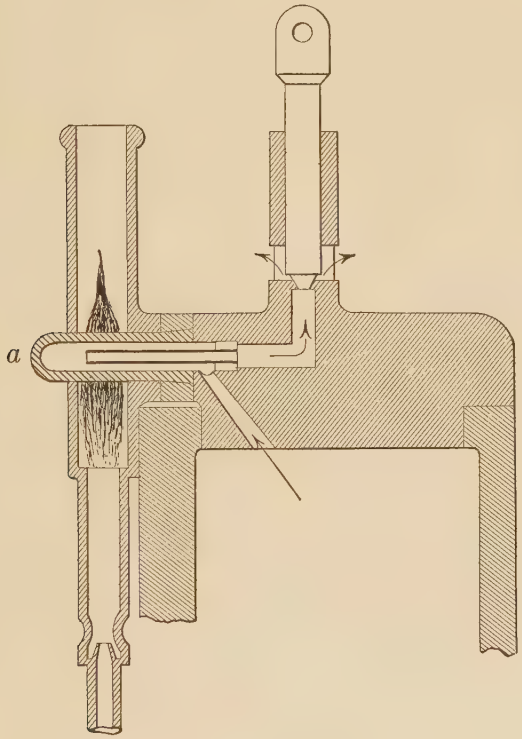


Fig. 29. — Typical Arrangement of Hot-tube.

system. Others are the hot-tube and the system in which ignition is due to high temperature of compression.

The Hot-tube. — A typical hot-tube arrangement is shown in Fig. 29. The tube *a* is made of porcelain or a nickel alloy, and communicates with the combustion chamber. It is heated, for starting, by a Bunsen burner or a gasoline torch. After the engine has been running for a short time, the tube will remain at a temperature high enough to fire the charge.

When the exhaust stroke has been completed, the tube will remain full of inert gases which will be at approximately atmospheric pressure. As the pressure builds up during compression, a small portion of the fresh charge is forced into the tube until finally it comes in contact with the part of the tube that is at or above the ignition temperature of the mixture. When this occurs, the flame strikes back along the tube and ignites the whole charge.

The **timing** can never be exact when a hot-tube is used. If the system described above, where the tube retains heat from the former explosion, is used, the ignition point will vary with the compression or the quality of the fuel. But, if the Bunsen burner is used to keep the tube hot, timing may be changed to a slight extent by moving the burner out towards the end of the tube for later, or in towards the combustion chamber for earlier, ignition.

The **advantages** of the hot-tube ignition are that it has no moving parts to wear out or get out of order and it is simple and is certain. However, it has the **disadvantage** of being low in efficiency, and, as has already been mentioned, it cannot be regulated with any degree of accuracy.

Electric Ignition. — There are many forms of electric ignition, and the scope of this book will not permit more than a superficial examination of a few of the more important types. Generally, the electric ignition system consists of a **circuit of wire,** or a combination of wire and the metal of the machine, a gap or **circuit-breaking device** inside the cylinder, a **commutator** or **distributing device** outside the cylinder and a **source** of **electric current.** Other devices used are a **switch, induction-coil** or **kick-coil** and **condenser.** The source of current is the **wet-cell, dry-cell, storage-battery** or **magneto.**

The systems may be classified as **low-** and **high-tension,** or **jump-spark** and **make-and-break.** The make-and-break system is usually used when low-tension current is used, and the jump-spark system with high-tension current.

Low-tension Ignition.— As the name implies, low-voltage current is used with this system, the source of which may be dry-batteries, storage-cells, generator or magneto. The tension is, on the average, from 5 to 6 **volts,** and the resistance is such that from 2 to 5 **amperes** flow when the circuit is closed.

In Fig. 30 is shown a general diagrammatic arrangement of a make-and-break low-tension ignition system. When the "electrodes" are in contact and the switch closed, the current flows through the circuit until the electrodes are separated. When they are suddenly separated, a spark passes between them, igniting the charge in the cylinder, if the charge is at the proper condition of mixture and compression. The passing of the spark is due to the action of the coil in the circuit. This coil is termed

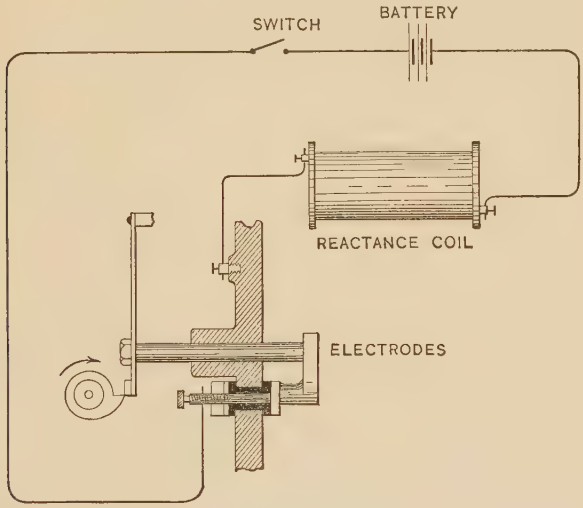


FIG. 30.—Diagram of Make-and-break Ignition System.

the "reactance," "kick" or "intensifying" coil. It consists of a number of turns of insulated wire wound on a soft-iron core. On account of the inductive action of such a coil a momentary increase of voltage is caused when the circuit is broken by the moving electrode.

This make-and-break apparatus is very simple electrically, but mechanically it is more complicated. The device used to give motion to the moving electrode must necessarily be more or less complicated in order to make it convenient to change the time of ignition. It is also troublesome to prevent leakage around the moving electrode where it enters the cylinder. Lubrication is difficult and it is hard to keep the electrodes cool enough to work efficiently. For high-speed engines special refinements are neces-

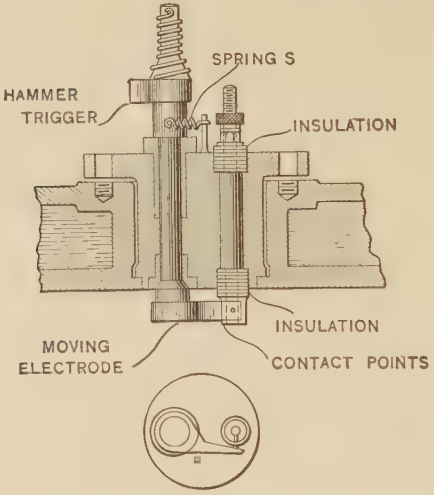


FIG. 31.—Make-and-break Igniter.

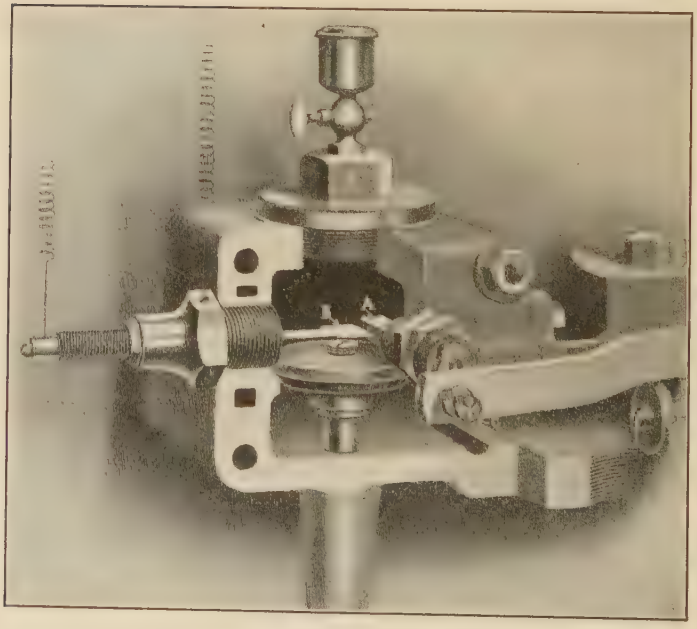


FIG. 32.—Foos "Wiper-contact" Igniter.

sary in order to make the inertia effects as small as possible, and to make the lost motion, due to wear, a minimum.

A typical **igniter** is shown in Fig. 31. The points on the electrodes are brought into contact some time before the spark is to be made. A push rod engages the "hammer trigger" which has no rigid connection with the moving electrode, being connected only through a spring. As the end of the push rod slips off the igniter cam it allows the electrodes to be pulled apart suddenly by the spring S, making the spark in the cylinder.

The make-and-break system described in the last paragraph is of the hammer type. Another type, the Wiper Contact Igniter, is shown in Fig. 32. This type is used by the Foos Gas Engine Company and is favored by them on the ground that the rubbing of one electrode over the other keeps both clean and free from carbon and other deposits, which may interfere, to a certain extent, with the proper working of the hammer type.

Make-and-break Mechanism.—One mechanical device used for moving the electrode is shown in Fig. 33. The cam C on the cam-shaft gradually moves the push rod P to the left. As this is done, the tappet T, which is hinged, passes over the hammer H without disturbing it. However, when the roller R drops off the high point of the cam, the spring S quickly forces the push rod to the right and the tappet now strikes H, forcing the electrodes apart in the combustion chamber, causing the spark. The current is timed so that it is passing through the electrodes only at the instant it is needed. As the electrodes are in contact a large percentage of the time, the current is shut off during this period.

In order to change the **time** of the ignition, the hand wheel W is supplied. It is evident that if the roller R be raised a certain distance, the spark will be earlier in proportion and if the roller be lowered, the ignition will be later. The wheel serves this purpose and to allow this motion the bearing B is pivoted.

The spring S_1 is inserted for flexibility. If a rigid rod were used in place of the spring, and the engine ran backwards for part of a revolution in stopping, as often occurs, the mechanism would be damaged. The spring prevents any damage as it allows the cam to force the roller down, extending the spring. As the roller is released, the spring brings the mechanism back to normal position.

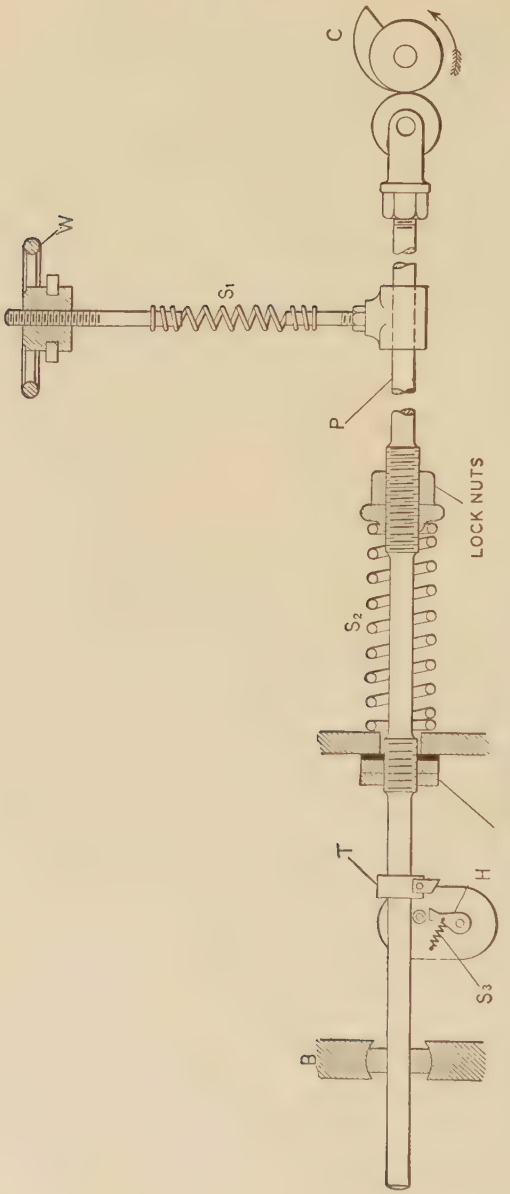


FIG. 33.—Make-and-break Mechanism.

High-tension Ignition.—A single electric spark, or series of sparks, jumping across a permanent break in the metallic circuit, and passing through the combustible mixture in the cylinder or an engine, is a means adopted to a very considerable extent for igniting the charge in an internal combustion engine. A high electromotive force, or tension, is required to force the spark across the gap. This is secured by use of the induction coil which changes a current of an ampere or two, with a tension of a few volts, into a current of much higher tension with a corresponding decrease in the current. The low-tension current is supplied by a dry-battery, storage-battery, magneto or small lighting gener-

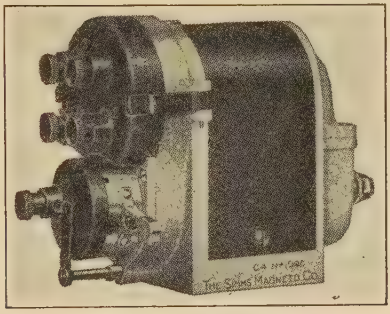


FIG. 34.—Simms High-tension Magneto.

ator. A "timer" is used in connection with the battery. The function of the timer is to close the battery circuit so that a current can flow from the battery through the induction coil at the proper instant to produce a spark in the combustion chamber.

The **generator,** the **timer,** the **induction-coil** and a **distributer** for directing the high-tension current to the different combustion chambers of a multicylinder motor are all sometimes brought together and embodied in a single piece of apparatus. This combination is called the high-tension magneto.

Figs. 34 and 35 show a Simms' high-tension magneto. It consists of a permanent horseshoe magnet to which are attached pole pieces which concentrate the lines of force across the pole faces. Rotating within this magnetic field is a shuttle-shaped armature which is wound with a few turns of heavy primary wire covered with many turns of a very fine secondary wire. The rotation of the armature causes a low-tension current to be generated in the

primary windings which is transformed into a high-tension current in the secondary winding precisely as in the induction coil described later on.

At the left-hand end of the armature a hard rubber spool is shown with a metal ring around it connected to the secondary winding. In contact with this metal ring is a carbon brush which transmits the high-tension current to the distributor shaft shown above and parallel to the armature.

The face of this shaft carries an insulated disc with a brass sector thereon attached to the secondary winding. This disc is

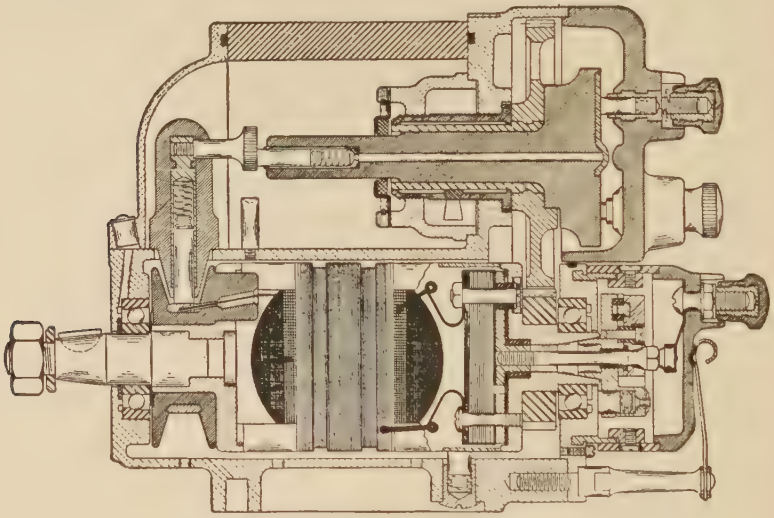


FIG. 35.—Section through Simms High-tension Magneto.

driven at half engine speed by gears from the armature shaft and in rotating delivers high-tension current to the terminals shown on the distributor cover and from these direct to the spark plugs.

At the extreme right-hand end of the armature shaft is a contact breaker consisting of a rotating bell crank which is actuated by two cams diametrically opposite carried in the contact breaker cover. These cams separate the contact points every time a spark is required. It is characteristic of the magneto that the current is always grounded except when a spark takes place. The laminated part shown to the right of the armature winding is the condenser which functions as described in the section on

induction coils. The high-tension magneto is widely used in automobiles wherever a self-contained ignition device is required and is also used to some extent on stationary engines.

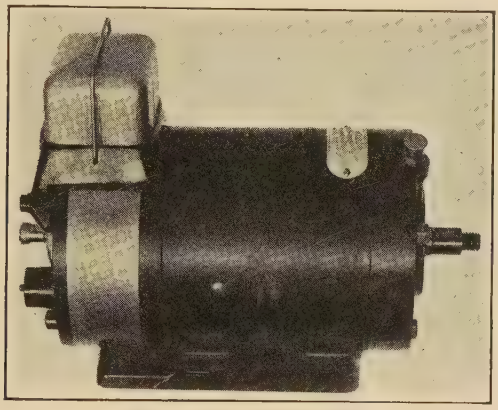


FIG. 36.—Delco Generator.

The universal use of electric lighting for automobiles has, however, caused a resuscitation of the high-tension coil ignition system. Since a generator and battery must be carried for the lighting system it becomes logical to use the same current for ignition purposes.

Fig. 36 shows a Delco generator used on motor cars for furnishing current for ignition, lighting and for charging the storage battery. The distributor used in connection with this generator for ignition purposes is shown in Fig. 37. Inside the distributor casing is a breaker mechanism for interrupting the primary current from the generator to produce a spark in the secondary circuit at the proper time. The points of this breaker are separated mechanically by a cam which has as many sides as the engine has cylinders. At the instant the contact points are separated by this cam the primary circuit through the ignition coil is interrupted. This induces a high-tension current into the secondary winding which in turn is distributed through the rotor and the distributor head for firing the cylinder under compression.

FIG. 37.—Delco Distributor.

The distributor shown is for use on a six-cylinder engine, the terminals shown in the form of a circle are for the wires leading

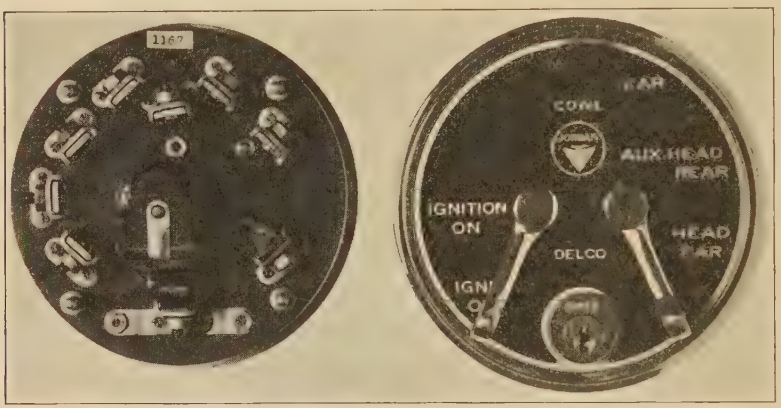


Fig. 38.—Delco Switch.

to the spark plugs, the one in the center of the circle is for the high tension from the induction coil which is usually located on the dash under the hood of the car. By shifting the spark lever on the steering wheel the distributor housing and contacts are shifted changing the relation of the lobes of the cam to the rubbing block on the breaker arm thereby advancing or retarding the spark.

Fig. 39.—Delco Starting Motor.

The generator is simply a two-pole, six- or twelve-volt shunt-wound dynamo with a third brush which controls the shunt field causing the generator to commence charging at low speeds, preventing the output from increasing unduly with increased speed and at higher speeds causing the output to be decreased. At speeds which are too low for the generator to begin charging or delivering current, the ignition current is furnished by the battery but at higher speeds the generator picks up the ignition load and also furnishes excess current which is used to charge the battery. The connections for this arrangement are contained

in one lever switch, the one shown at the left in Fig. 38. The switch lever at the left controls the lights in various combinations.

Ignition and starting being intimately related, a Delco engine starter with Bendix drive is shown in Fig. 39. The starter is a small electric motor with a square thread mounted on the end of the armature shaft. When the starting switch is closed the rapid acceleration of the armature shaft causes the starting pinion to move along the thread and mesh with a gear cut on the outside of the flywheel thus rotating the same at a speed of approximately 100 revolutions per minute. Immediately the engine fires the flywheel gear speeds up thus increasing the speed of the starter pinion which is in mesh with it. This increased speed of the pinion brings into play the shift mechanism which draws the pinion out of mesh. The starting switch mentioned above connects the starting motor direct with the battery and the current used does not pass through the switch shown in Fig. 38.

Fig. 40.—Spark Plug for High-tension Ignition.

Spark Plugs.—The jump-spark igniter is, in almost all cases, made up of a central metal wire surrounded by a thick tube of insulating material which, in turn, fits into a hollow metal plug threaded on the outside so as to screw into a threaded hole in the wall of the cylinder of the engine or into some part that fits into the cylinder wall. The insulating material is generally either **porcelain** or **mica.** The former is used in one tubular piece, and the latter is made up of numerous discs perforated for a central wire and placed side by side over it. **Lava** is used for insulation to a limited extent. When the porcelain is used, tight joints are made between the porcelain and the central wire and between the porcelain and the outer bushing by the use of **asbestos fibre packing** or **soft copper washers.** In general practice either the central wire of the plug terminates in a wire of smaller diameter brought near the outer shell, or a small wire is fastened to the outer shell and brought near the enlarged end of the central wire. The gap thus left in either case is called the "spark gap." Its width is usually about $\frac{1}{32}$ inch. The size of the threads of

the American made plugs corresponds to the standard half-inch pipe tap or $\frac{7}{8}''$ diameter, 18 threads per inch. French plugs are smaller at the threaded part of the outer shell and approximately the same dimensions elsewhere.

Timers for High-tension Ignition.—When a battery is used to supply current for the **high-tension** jump-spark ignition, a **timer** is placed in the battery circuit to close it at the instant a spark is required. The timer thus controls the time of flow of the low-tension current. One of the principal parts of the timer is stationary, except for a slight adjusting movement, and the other rotates. These are electrically insulated from each other. As the rotor revolves, a metal contact piece on it comes against the metal of the stationary part at intervals and closes the circuit. In the more common type, used where there is a separate induction-coil for each cylinder, there is a contact piece on the stationary part for each coil. The contact pieces are placed around the circular path of the rotating piece in order to close the circuit at the time required by each cylinder of the motor. The stationary part is adjustable to a certain degree by rotary motion around the rotor shaft, so that the time of closing the circuit, and hence the explosions, can be varied to suit the needs of the motor.

In the best modern designs the stationary part generally consists of a ring of **wood fibre** supported on a metal part that is bored to receive the shaft of the rotor. The contact points of the stationary part are attached to the insulating ring so as to be insulated from each other and from the shaft of the rotor. The rotor has only one contact piece, and in some designs it has rigid metallic connection with the rotor shaft; in others it is insulated from the shaft, but is permanently connected by rubbing or rolling contact with the metal ring that is part of the stationary member of the timer, and is electrically connected to the metal of the motor. The contacts are pressed together by the action of a spring. The moving parts are packed in soft grease or lubricated with plenty of oil.

In automobile practice the frequent movement of the adjustable piece in advancing and retarding the spark, is liable to break the wires that lead from the contact pieces to other parts of the apparatus. In order to prevent this trouble the adjustable part is surrounded by a case that is truly stationary with regard to

the motor frame, and the leading-out wires are connected to binding posts on the casing. The electrical connections between the case and the adjustable part are made by sliding contacts.

The speed of the rotor of the timer is half that of the crank-shaft in four-cycle engines and the same as the crank-shaft in two-cycle engines.

Induction-coils.—The induction-coil used for **high-tension** ignition in gas engine and motor practice has a central core of very soft, small iron wires arranged in a bundle. Insulating material in the form of a tube covers the core. Comparatively coarse copper wire is wound around the insulating tube in the form of a solenoid coil of a few layers of wire with several turns in each layer. This is the "low-tension," "primary" or "battery" coil. The turns of wire are insulated from each other by using a wire with an insulating cover or by carefully winding bare wire with sheet insulation between the layers, and then filling the spaces in between the wires with paraffine. One end of the primary coil is attached to a binding post for receiving a battery wire, and the other end connects to a device for interrupting the current.

The interrupter has a thin flat spring (vibrator, trembler) that is rigidly held at one end so that a metal contact point near the free end is pressed against the mating point. The metal parts to which the two contact points are attached are electrically insulated from each other when the points are separated. The second wire from the battery is connected to the part of the interrupter that is insulated from the side to which the primary-coil wire is connected. The free end of the spring has attached to it a disc of soft iron that is held just opposite one end of a soft-iron core of the coil and a short distance from it. When a current of electricity is passed through the coil it magnetizes the core which then attracts the metal disc and draws it, together with the free end of the spring, toward the core. The contact points are thus separated and the current interrupted. The core then quickly loses its magnetism, and the elasticity of the spring brings the points together again so that the current again passes through the coil. This operation is repeated and continued as long as sufficient current is supplied.

A second coil, the **secondary or high-tension,** is wound over the first. It is of exceedingly fine wire and has a great many

turns. The turns and layers are separated from each other as in the primary coil. One end of the high-tension coil is connected to the same binding post as the end of the low-tension coil. The other end of the high-tension coil is attached to a binding post of its own. The apparatus thus has three binding posts for receiving wires from the outside. One terminal is at the battery side of the interrupter; another, which may be called the intermediate terminal, is between the ends of the high- and low-tension coils; and the third terminal is at the remaining end of the high-tension coil.

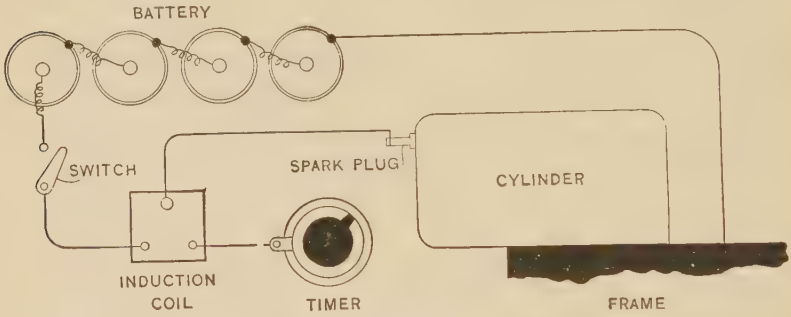


Fig. 41.—Wiring Diagram for High-tension System.

When the **battery current** stops flowing through the **primary winding** it induces a current of **high-pressure** and small volume, or amperage, in the **secondary winding.** For ignition purposes the tension of the secondary current should be high enough to give a spark across a one-quarter-inch air gap.

An electric **condenser** is used in connection with an induction-coil. Its function is to strengthen the action of the coil and prevent the contact points of the interrupter from fusing. When the contact points of the interrupter are separated there is a tendency for the battery current to keep flowing in an arc across the gap formed. The magnetic core, acting inductively on the primary coil, has a tendency to maintain the arc. The condenser counteracts the combined effort by receiving and storing the electrical energy and thus breaking down the arc quickly. The energy stored in the condenser is probably discharged back through the primary circuit immediately after the primary circuit is broken, thus further increasing the inductive action and the strength of the spark.

The **condenser** is made up of sheets of tin foil and paraffined paper laid together alternately, the paper insulating the sheets of foil from each other. Alternate sheets of foil are connected together electrically to form one pole of the condenser, and the remaining sheets are likewise connected together to form the other pole. One of these poles is connected to the battery side of the interrupter, and the other to the primary coil side.

American made induction-coils for ignition purposes are generally wound to operate on 6 or 7 volts. Most of the European

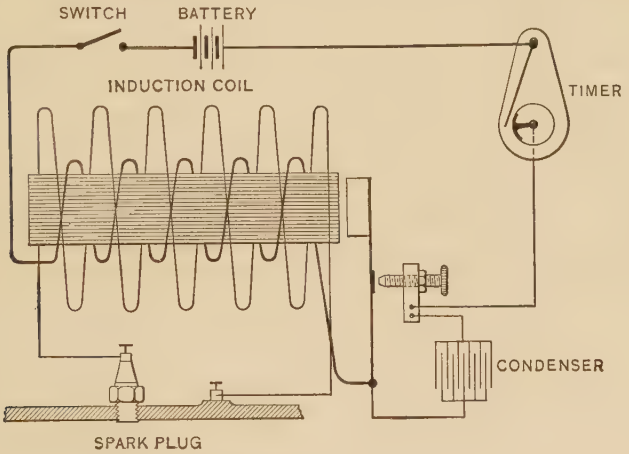


FIG. 42.—Diagram of High-tension Ignition System with Condenser.

made coils are for a voltage on the primary of 4 volts. Higher voltages than these should not be used on account of trouble in melting contact points and breaking down the insulation of the winding.

The wires and connections of a high-tension ignition system are shown in Fig. 41 In this case the condenser and induction-coil are contained in one case and a minimum amount of wire is used, the wires being grounded to the metal of the engine wherever possible. In Fig. 42 the wiring is shown with a separate condenser connected across the vibrator points. Here the only "ground" is at the spark plug itself.

CHAPTER IX

CARBURETION AND CARBURETORS—TYPES OF CARBURETORS

Carburetion.—Carburetion is the process of making a combustible mixture of fuel vapor and air by passing air, cold or hot, over or through a spray of the volatile liquid fuel. The amount of fuel vapor that will be carried by the air depends on the vapor tension of the fuel. When the vapor tension of the fuel is, under ordinary atmospheric conditions, much less than that of air, it may happen that the ratio of fuel to air in a saturated mixture is too small to form a good explosive mixture. On the other hand, fuels with high vapor pressures form saturated mixtures with air at ordinary temperatures and pressures that are too rich to burn and must be diluted before they will explode in a gas engine cylinder.

In Chapter V it was pointed out why certain liquid fuels form explosive mixtures with air very readily at normal temperatures and others must be heated before such mixtures can be formed. The **gasoline** which is used today in automobile motors and stationary gasoline engines has so much lower vapor pressure compared to that used ten years ago that the matter of carburetion has become very important; and at the same time it is a problem that is very difficult to solve in a way that is entirely satisfactory. Formerly, carburetors were not water-jacketed, nor was any special provision made for supplying warm air to the carburetor. Today both these measures are resorted to in order to give good results, and some carbureters are jacketed with exhaust gases.

When it is attempted to use **kerosene** or **alcohol,** the matter of preheating the fuel and air becomes still more important. With kerosene it is necessary to supply heat on account of its low vapor pressure of saturation at ordinary temperatures. With alcohol the same difficulty presents itself, and in addition, the water which is always present in commercial alcohol gives trouble.

Kerosene cannot, in general, be used with a gasoline carburetor even if provision is made for jacketing with hot water or exhaust gases and supplying hot air. It is true that almost any gasoline engine which has been operated long enough to have reached a constant temperature can be made to continue in operation on a kerosene and air mixture formed in a gasoline carbureter. The operation will, however, be more or less uncertain and it will, in general, be only a matter of a few days before the interior of the combustion chamber is covered with a deposit of carbon and the piston rings are gummed tight in their grooves.

Kerosene, being less volatile than gasoline at ordinary temperatures and pressures, is not thoroughly vaporized in the ordinary gasoline carburetor, but some of the fuel is carried over into the cylinder in the liquid form. Part of this liquid will remain in that state until combustion has started. When this occurs the high temperature will tend to vaporize the excess liquid, but there is also another process known as "cracking" that goes on simultaneously. Cracking consists in breaking up the molecules of the liquid kerosene, some of the resultant products being finely divided carbon and heavy viscous or tarry liquids. It is this carbon and these heavy liquids that cause the trouble previously referred to by collecting in the cylinder.

The more the kerosene is broken up, that is, the more perfectly it is atomized or sprayed into the air, the less chance will there be for the collection of the liquid on cold surfaces and the more rapid will be the vaporization in the cylinder of the engine. Fine spraying is therefore advantageous, and any jet carbureter will work best when the pressure available for causing the flow is so great that the needle valve must be almost entirely closed.

It has also been found by experience that if a certain amount of **water vapor** or a finely divided water spray is mixed with the kerosene-air mixture, conditions are materially improved. No really satisfactory explanation of the full action of water vapor has yet been offered, but it seems probable that it is very complicated and of a physical as well as chemical nature.

The use of water makes possible the use of high compression without danger of pre-ignition and this makes possible the attainment of high efficiency. This is probably due to the fact that water absorbs heat during compression, thus keeping the temperature below that which would cause spontaneous ignition, and

there may also be some sort of chemical action whereby the presence of the water prevents the breaking up of, or the combination of, certain of the constituents of the fuel to form compounds that have very low ignition temperatures. The presence of water also tends to prevent the deposit of carbon in the cylinder and to make combustion more perfect, so that the exhaust is less smoky.

Types of Carburetors.—When a high grade of **gasoline** is used for fuel, a perfectly satisfactory mixture of air and vapor can be made very readily and at ordinary temperatures by simply injecting the liquid into the air as it passes on its way to the cylinder. Devices arranged thus to inject fuel into the air may be classified under the head of jet carburetors.

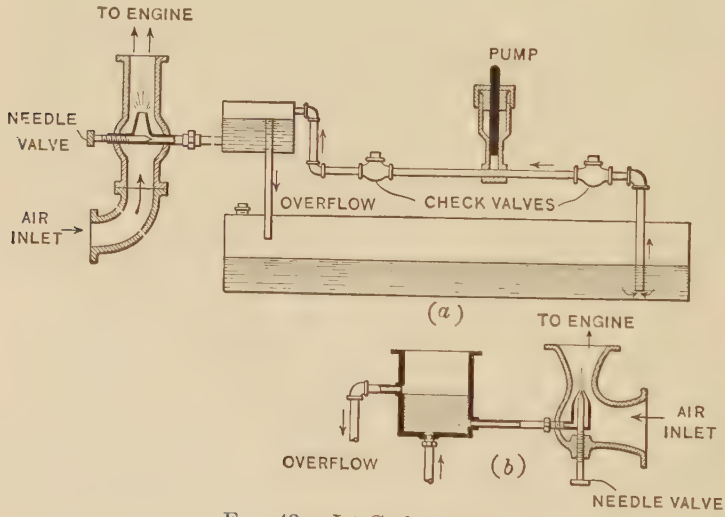


FIG. 43.—Jet Carburetors.

Jet Carburetors.—In this form of carburetor the nozzle is supplied from a small reservoir in which the fuel is maintained at a fixed level slightly below that of the nozzle so that normally there will be no flow from the nozzle. During the suction stroke of the engine a flow is established as described below. The level in the reservoir is maintained in several ways. One method of keeping the level constant is by means of a float and needle valve. This method has given rise to a particular type of carbureter known as the "float-feed." This type will be explained later. A simple method of maintaining a constant level is shown in Fig. 43. An

overflow from the reservoir back to the fuel tank carries away any surplus fuel. In this type a pump must be used to lift the fuel from the storage tank to the constant-level reservoir.

It will be noticed that in both (*a*) and (*b*), Fig. 43, the nozzle enters the air pipe leading to the inlet valve of the engine, or is centered in that pipe. The diameter of the suction or air pipe is often reduced as shown at (*b*), thus forming a Venturi tube. This increases the velocity of the inlet air at that point and assists in the mechanical separation of the particles of gasoline. During the suction stroke of the engine, the pressure within the air pipe is reduced below that of the atmosphere, and the air rushing past the nozzle creates an additional slight pressure drop immediately opposite the end of the latter. As a result of this double reduction in pressure opposite the end of the nozzle, a small jet of gasoline is discharged into the air current by atmospheric pressure on the surface of the liquid in the reservoir. With a Venturi passage a considerable decrease in pressure may be obtained at the nozzle and the pressure built up again to nearly atmospheric conditions at the inlet valve.

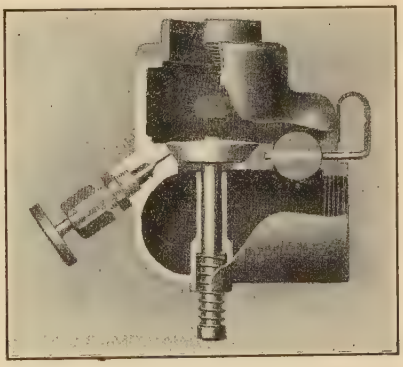


FIG. 44.—Carburetor Valve.

The quantity of gasoline thus discharged can be regulated by means of the needle valve shown in the figure. When hit-and-miss governing is used it is obvious that no fuel need be drawn out of the nozzle and wasted during the missed cycles. When throttling governing is used, the throttle valve may be placed on the engine side of the carburetor, thus controlling the quantity and velocity of the air through the latter and indirectly controlling the amount of gasoline discharged into that air.

Another simple form of carburetor, known as the carburetor valve, is shown in Fig. 44. The light disc acting as a poppet valve is inserted in the air pipe and closes off the supply of the air and the gasoline as shown. During the suction stroke this valve is raised by the atmospheric pressure, and, as the gasoline

outlet is uncovered, a jet spurts into the air as it passes on its way to the cylinder. The gasoline may be supplied from a constant head reservoir with a level slightly below the discharge point, or preferably under slight pressure, pressure to be from natural head or by air pressure on the supply tank.

Carbureting valves do not form a very good mixture of fuel vapor and air and are consequently used only on small engines. They are used frequently on two-cycle engines having crank-case compression and are fairly successful as the churning action in the crank-case has a tendency to complete the thorough mixing started by the valve. The two properties which recommend this valve for use on small engines are low cost and simplicity.

Float Carburetors.—The carburetor used on practically all multicylinder gasoline-engines and on all automobile and marine

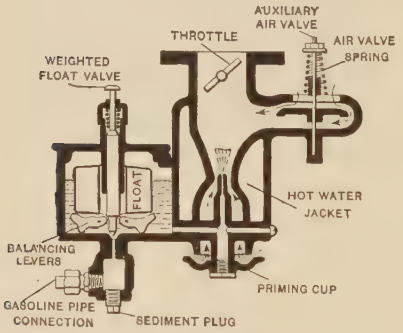


FIG. 45.—Carburetor with Eccentric Float.

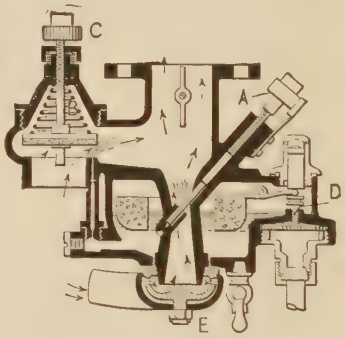


FIG. 46.—Carburetor with Concentric Float.

motors is of the jet type in which the level in the small reservoir near the jet is kept constant by means of a float and needle valve. The float is made of sheet copper or cork, the latter material being the favorite.

The position of the float relative to the spray jet has resulted in two sub-types of carburetors. If the float is in a separate and distinct chamber, at one side of the jet, the carbureter is of the "eccentric float" type, but if the float chamber surrounds the jet and air passage, the carbureter is of the "concentric float" type. These two types are illustrated in Figs. 45 and 46 respectively.

Gasoline Adjustment.—The ratio of gasoline to air in a mixture that is drawn into a gasoline engine must be readily changed to

suit varying conditions due to atmospheric changes and also variations in the quality of the gasoline. This adjustment is usually made by the needle valve in the spray nozzle mentioned above. Other methods have been used but the needle valve seems to be the most convenient and the most satisfactory.

Air Adjustment.—The air that passes around the jet of gasoline becomes saturated with the fuel, especially if this air be warm, as it is when taken from a jacket surrounding the exhaust

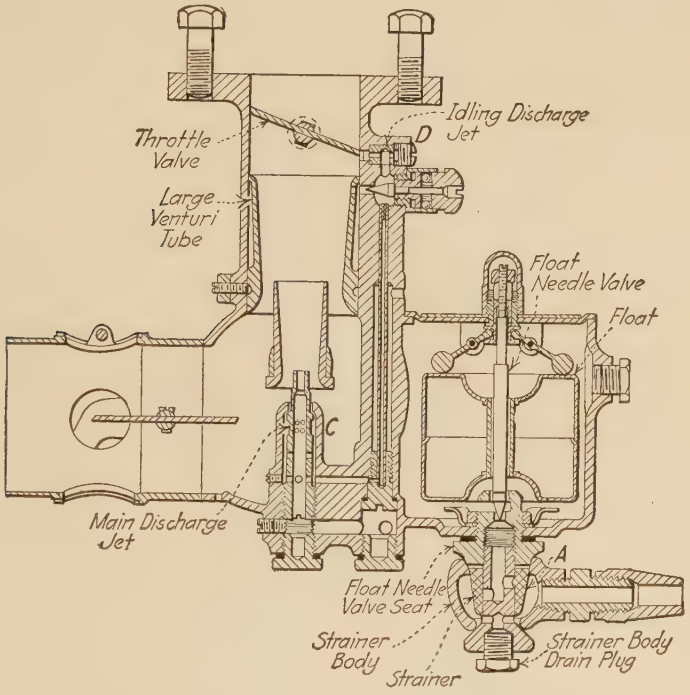


FIG. 47.—Stromberg Carburetor. Section through Float Chamber and Main Jet.

pipe. Since this saturated air is too rich to form an explosive mixture, auxiliary air is drawn in to dilute the charge between the carburetor and the engine. The quantity of auxiliary air is regulated to suit the existing conditions, by means of a butterfly valve or an automatic lift valve. It is the difference in pressure between the fuel and the atmosphere which causes both the air and the fuel to flow. As the engine speeds up this pressure differ-

ence increases and hence the flow of both air and fuel increases. Unfortunately the flow of liquid fuel increases faster than that of the air, causing a richer mixture as the speed increases. For this reason the auxiliary air valve is required and it must allow more and more air to flow as speed increases.

The type O Stromberg carburetor shown in Figs. 47*A*, 47*B* and 48 is typical of the latest design in plain tube carburetors.

The principle of a plain tube carbureter is that of having both the air openings and fuel jets of fixed size, the air supply creating a suction according to the rate of flow which determines automatically the correct amount of fuel for all loads and speeds.

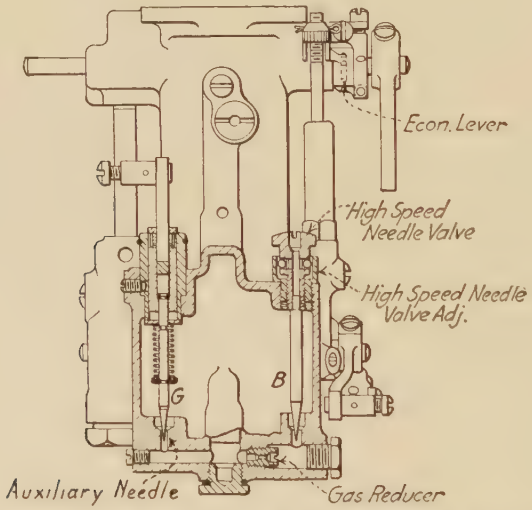


FIG. 47*A*.—Stromberg Carburetor. Section Showing High Speed and and Auxiliary Needles.

The fuel after leaving the dirt trap or strainer *A* enters the float chamber where a constant level is maintained.

From the float chamber the fuel is drawn by suction past the metering opening of the high speed needle *B*, to the main discharge jet *C*, or to the idle jet *D* according to whether the throttle is opened or closed.

The main air passages are shaped to give a free flow, but of high velocity past the fuel jets. The air passages used are known as venturi tubes.

The special application of a double or compound venturi tube is used to obtain a high velocity at the fuel jet without undue restriction, and in this manner secure the highest degree of atomizing action at the fuel jet at low air velocity, such as takes place when a motor is pulling with wide open throttle at low engine speed.

In carburetor designs of the earlier days, the mixture strength was maintained by air valves, metering needles, compensating jets and the like, but they were at best unsatisfactory and unreliable. In this new design of carburetor the mixture regulation is obtained automatically and with great accuracy by the simple

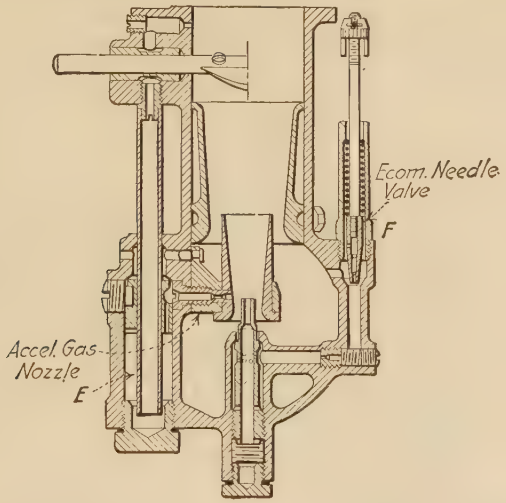


Fig. 47*B*.—Stromberg Carburetor. Section Showing Economy Needle Valve and Accelerating Gas Nozzle.

use of an air-bled jet, which consists of correctly located small holes in the side of the fuel jet between the metering orifice and the discharge point, or point of suction. Air drawn in through these holes in the form of tiny bubbles breaks up the fuel discharge and frees it from the retarding action of surface tension at low suctions, and restricts the gasoline flow at high suctions, thus giving perfect compensation or maintenance of the correct air fuel ratio at all speeds.

In general a carburetor adjustment giving an economical mixture lags in response to a quick opening of the throttle, even in

well-heated intake manifolds. On the other hand adjustments that answer promptly to the opening of the throttle, show considerable increase in fuel consumption.

The reason for the lag on acceleration is brought about by the condition in which the fuel exists in the manifold when the throttle is opened quickly. The sudden drop in velocity and pressure in the manifold causes only a small percentage of the fuel to

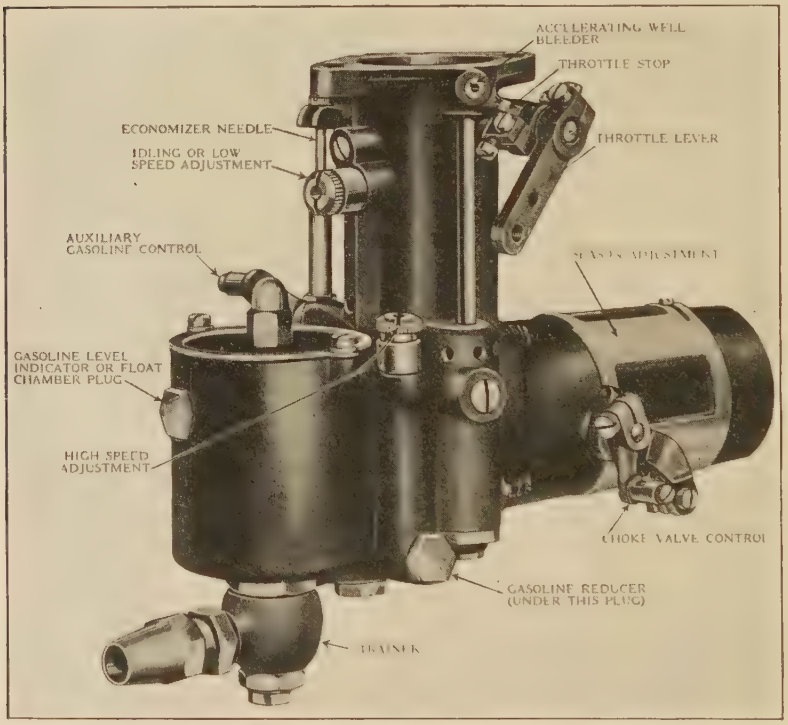


Fig. 48.—Stromberg Type-O Carburetor.

vaporize and the balance of heavy particles is deposited on the manifold walls and consequently retarded in its flow to the cylinder.

It is, therefore, necessary to supply a small amount of extra fuel to supply the necessary volatile particles to make a proper mixture.

The accelerating well *E* furnishes this extra amount of gasoline, automatically, when necessary and thus permits of an economical

setting of the carburetor without the disadvantages of poor performance and lag on acceleration.

The fuel discharge from the accelerating well is controlled by the vacuum in the manifold and thus furnishes extra fuel, only at a time when it is required by the motor.

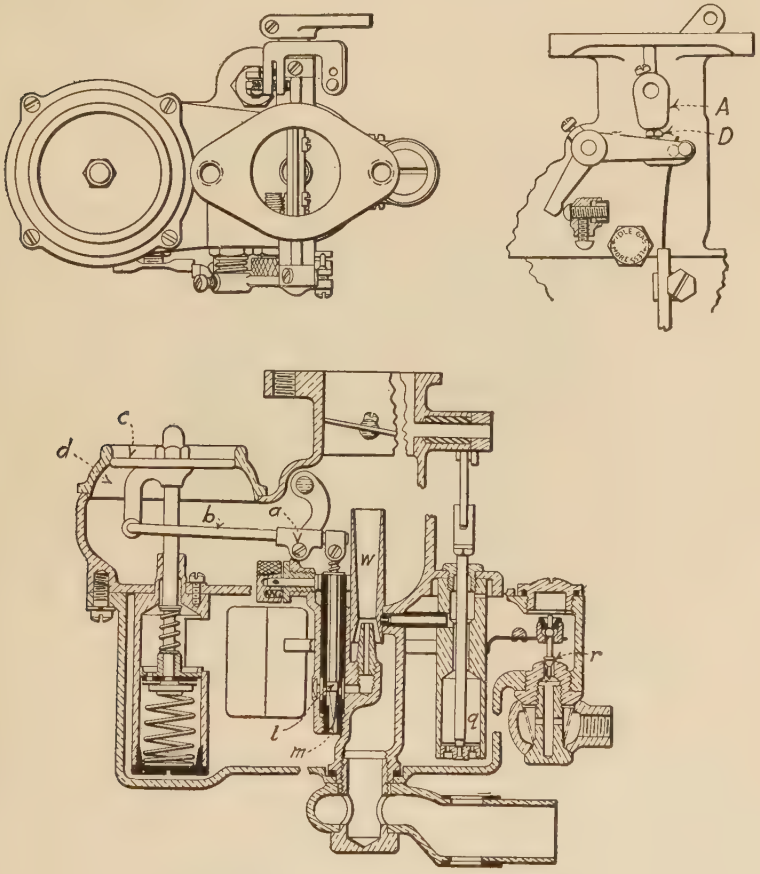


FIG. 49.—Schebler Model S Carburetor.

The economizer needle *F* at partly open throttle or light loads makes it possible to run with a leaner mixture or a higher air fuel ratio than at wide open throttle by governing the amount of air that is bled into the main fuel stream in the jet at part throttle. This leaning out at part throttle makes it possible to secure

maximum fuel economy with a setting that gives maximum power at wide open throttle.

The auxiliary gasoline control *G* renders immediate driving possible with a cold motor, *i.e.*, just after starting, at which time it is necessary to supply a somewhat richer mixture owing to the low volatility of fuel which now prevails all over the American Continent.

An external view of this carburetor is shown in Fig. 48.

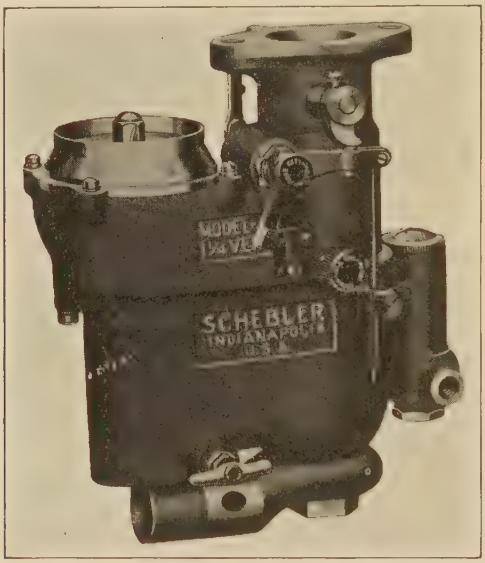


FIG. 50.—Schebler Model-S Carburetor.

The model "S" Schebler Carburetor, shown in Figs. 49 and 50, is an air valve carburetor having two air inlets. The fixed air opening is through the venturi *w* and is very small, furnishing barely enough air to enable an engine to idle slowly.

The auxiliary air valve *c* is closed for idling and wide open at full power, thus allowing the carburetor to handle a large volume of air with a small suction.

The gasoline enters the float chamber through the valve *r* and after leaving it is metered by the gasoline metering pin *m* at the point *l*. It then passes on into the air stream through the venturi *w*.

The correct fuel-to-air ratio is maintained by the proportioning lever *b* which rotates about the fulcrum point *a*, thus giving to the gasoline needle *m* a definite part of the air valve motion. Any desired variation in the richness of the mixture at different rates of flow can be obtained by changing the shape of the inner surface of the air funnel *d*, thus controlling the area of the air opening for any given movement of the air valve. It will also be seen

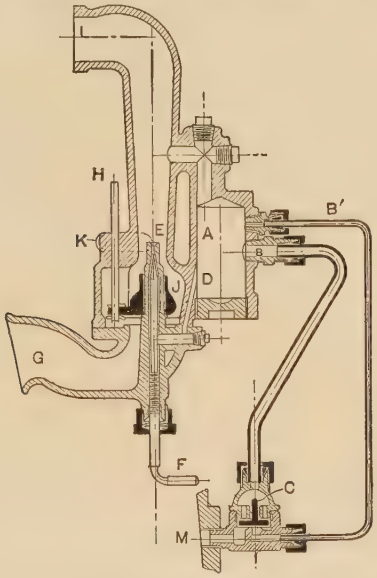


FIG. 51.—Mixer Used on Stationary Gasoline Engines. Seager Gas-engine Works.

that when the throttle valve is wide open the cam *A* depresses the pin *D* thus moving the fulcrum point *a* over towards the air valve and leaving the metering pin *l*, this causing an increase in the richness of the mixture so that full power can be obtained at wide open throttle without sacrificing economy at part throttle.

The accelerating pump *q* comes into operation when the throttle is suddenly opened thus giving the necessary increase in richness of mixture to prevent the misfiring which would otherwise occur.

Mixer—Stationary Engine.—The mixer used on gasoline engines built by the Seager Gas-engine Works of Lansing, Michigan, is shown in Fig. 51. The pipe *L* is connected to the inlet valve

chamber, and air for combustion is drawn up from G. This produces a suction in chamber A which tends to draw up gasoline from the tank at M. The check valve C lifts during the suction stroke, closing pipe B but opening pipe B', thus allowing the gasoline to be drawn up into the chamber A. Pipe B acts as an overflow, down which the surplus gasoline may flow during any stroke of the engine except the suction.

Gasoline flows from chamber A to the nozzle E, which is about $\frac{1}{8}$ inch higher than the overflow. This spray nozzle is adjusted by the needle valve F. In order to enrich the mixture for starting, the weight J is lifted by the rod H, closing off part of the air in the Venturi throat. The spring K holds H with enough force so that J will not return by gravity.

CHAPTER X

GOVERNING — THEORY OF WATT GOVERNOR — SYSTEMS OF GOVERNING — TYPES OF GOVERNORS

Governing. — Governing is the process of changing the power generated in the cylinder to correspond to the changes in load. The governor, the apparatus that effects this change in power, should be beyond criticism as to reliability, for on the governor depends the success of the engine. The governor must be sensitive to speed changes and must vary the power immediately to suit the speed changes. The operation of the governor must be smooth and without fluctuation.

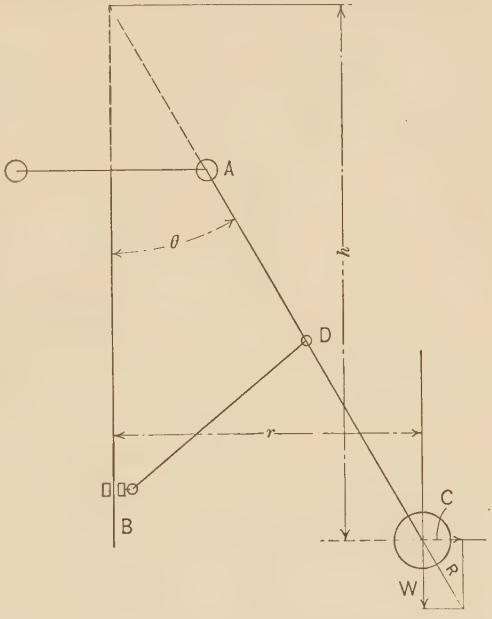


Fig. 52. — Diagram of Watt Governor.

The governor proper, on a large percentage of engines, consists of two balls or weights, one on either side of a spindle, which revolve at a high speed. These balls or weights are connected by a system of links to a sliding collar on the spindle. As the spindle and balls revolve, the centrifugal force of the balls projects them in a radial direction, and by this movement the sliding collar is lifted. To this collar is also attached the mechanism for changing the power in the cylinder.

This simple governor, called the Watt governor, is illustrated in Fig. 52, which shows only one of the balls. For purposes of calculation it is generally assumed that the weight of the two balls

is concentrated in one, which assumption does not affect the results of the calculation. The balls rotate about the spindle with an angular velocity proportional to the speed of the engine. If the engine speed increases the balls move outward about A and the collar B is lifted. Since the link AD can turn freely about A, then, neglecting friction, it is necessary for equilibrium that the resultant of the weight W and the centrifugal force of the ball (that of the arm being neglected), should act in the direction of the link. If we take W vertical, then draw a horizontal line to meet the direction of the link and complete the parallelogram, C is the centrifugal force for that position of the governor. If the governor is revolving at N revolutions per minute, and ω is the angular velocity in radians per second,

$$\omega = \frac{2\pi N}{60},$$

$$C = \frac{W}{g}\omega^2 r = \frac{W}{g}\frac{4\pi^2 N^2 r}{60^2} = \frac{WN^2 r}{2940},$$

$$\frac{h}{r} = \frac{W}{C},$$

$$h = \frac{2940}{N^2} \text{ feet} = \frac{35{,}280}{N^2} \text{ inches},$$

$$N = \frac{54.2}{\sqrt{h}} (h \text{ in feet}) = \frac{187.8}{\sqrt{h}} (h \text{ in inches}).$$

If h_2 and h_1 are the heights to which the governor is limited by the stops which define the travel of the sleeve, then

$$N_2 - N_1 = 54.2\left(\frac{1}{\sqrt{h_2}} - \frac{1}{\sqrt{h_1}}\right)$$

is the range in speed within which the governor can exercise control. The mean speed is $N = \frac{N_2 + N_1}{2}$ and the variation from mean speed within the range is $\frac{\frac{1}{2}(N_2 - N_1)}{N}$ or $\frac{N_2 - N_1}{N_2 + N_1}$ expressed as a fraction of the mean speed or $100\,\frac{(N_2 - N_1)}{(N_2 + N_1)}$ as a percentage of the mean speed.

For ordinary cases the movement of the balls of a Watt governor corresponding to a given variation in speed is small, and the governor is comparatively insensitive. Thus, if the normal speed

of the governor is 100 and a variation of 2 per cent from the mean is allowed, so that the maximum and minimum speeds are 102 and 98, the corresponding heights will be

Revolutions per minute =	102	100	98
h =	3.39	3.53	3.67 inches
Differences		0.14	0.14

The Watt governor is stable in the sense that there is a definite position corresponding to any speed within its range. If the speed increases, C increases and the balls move outwards a definite distance, satisfying the condition that $\tan\theta = \frac{C}{W}$. In order that the governor may completely control the gas supply, the mechanism between the sleeve and the gas-valve must be so arranged that, as the sleeve moves from the lower to the upper stop, the valve will move from full open to shut.

Systems of Governing. — The governor may regulate the speed in various ways. One of the most familiar for small engines is the **hit-and-miss** system, in which explosions are discontinued when the load lightens and the engine speeds up. The second, in importance, is the **throttling** system in which the mixture is throttled between the carburetor or mixing valve and the inlet-valve. A third system is the **constant quantity** in which the proportion of gas and air are changed by the governor, and there is no throttling effect. Another system, which is very like the throttling, is the **constant quality,** in which the proportion of gas and air are unchanged, but the quantity is varied to suit the load. This result may be obtained by other methods than throttling.

Hit-and-miss Governing. — It has been stated elsewhere in this book that gas engines are not efficient at light loads. The reason is this: The cylinder volume is designed to give a certain maximum power and to generate this power the cylinder must be filled with gas and air in the proper ratio to secure best results. When this cylinderful of mixture is compressed into the clearance space, the pressure attained should be the proper pressure for obtaining complete combustion. If any other conditions exist, that is, if less than a cylinderful of mixture is drawn in and compressed, the power generated will be less but efficiency will be lowered, as the best conditions, those for which the engine was designed, do not prevail.

If a single-cylinder, single-acting, four-cycle engine with hit-and-miss governing be designed for 230 revolutions per minute, the maximum number of explosions would be 115 per minute. Allowing 15 per cent overload, the number of explosions at the rated load would be 100, at $\frac{3}{4}$ load 75, $\frac{1}{2}$ load 50, etc. All the explosions occur when a full charge is in the cylinder, hence combustion takes place under the best conditions, and efficiency is high, even at light loads.

There are several methods by which an engine can be made to miss an explosion; some are good and others are not. The simplest method is to break the ignition circuit for the required number of strokes, but this is not a good way as the very important advantage of the hit-and-miss governing, that of high efficiency at light loads, is lost. Even if the spark is cut out fuel is drawn into the cylinder discharged into the exhaust unburned. This is a source of danger as this unburned fuel often will explode when brought into contact with the hot muffler or exhaust pipe.

Another method, the one generally in use, is to allow the inlet-valve to remain closed during the suction stroke. If this valve be opened mechanically, the usual method is to have the cam-roller connected to the governor by a system of links. As the speed goes above the normal, the governor moves the roller out of line with the cam and the valve remains closed for that stroke or until the speed of the engine drops sufficiently to allow the roller and cam to engage again.

When the above method is used, some means should be provided to allow air to be drawn into the cylinder during the suction stroke. In one make of engine two inlet valves are used, one for air and one for gas, but both are opened by the same cam. The spring on the air valve is light enough, however, so that when the governor throws the roller free from the cam, the air valve opens, due to the partial vacuum in the cylinder, while the gas valve remains closed, owing to the stiffer spring on that valve.

Where an automatic valve is used, that is, one that opens due to the decreased pressure in the cylinder during the suction stroke, it is not possible to use the apparatus just described. Instead, there is an auxiliary cam next to the exhaust valve cam, which, in normal operation, is not in use. When the speed increases above normal, the governor moves the exhaust-roller so that it engages this auxiliary cam, which then opens the exhaust valve during the

suction stroke as well as the exhaust stroke. Since the exhaust valve is held open, there is no vacuum produced in the cylinder during the suction stroke, no charge enters and an explosion is missed.

One disadvantage of this type of governor mechanism is that long exhaust pipes may produce a partial vacuum in the cylinder due to the inertia of the column of gases rushing out of the pipe. The vacuum thus produced may be sufficient to open the inlet-valve slightly so that a small amount of fuel is let in and discharged into the exhaust without burning. Naturally, the fuel thus lost helps to reduce the efficiency of the engine.

Hit-and-miss Mechanisms. — Mechanisms used in connection with hit-and-miss operation are plentiful in number and vary widely in construction. Those using a fly-ball governor to move the parts are very simple. There is usually a trigger, called the

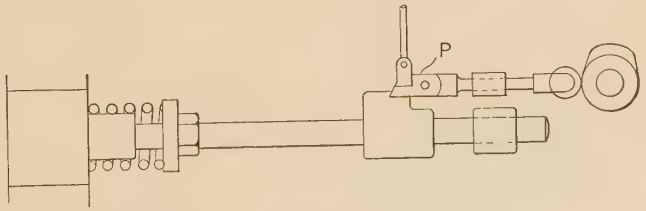


FIG. 53. — Hit-and-miss Mechanism Used in Connection with Watt Governor.

pick-blade, which is under the control of the governor as regards position. This pick-blade is moved by the cam and taps the valve rod if the valve is to be opened. If the valve is to remain closed, the governor moves the pick-blade out of the line of action. Fig. 53 illustrates this type.

Fig. 54 illustrates the hit-and-miss governor mechanism used on the smaller engines built by the Seager Gas-engine Works. The cam *C* operates the exhaust valve stem *D* through the bell crank. Attached to the bell crank is the arm carrying the notch *N*, which oscillates with the bell crank, moving to the right as the exhaust valve opens. The trigger *A* and pick-blade *B* form a bell crank, pivoted near *E*. As the governor weight *W* revolves with the flywheel, the finger *F* forces *A* to the left and hence lifts *B*, so that when the speed is normal, *B* cannot engage in *N*. However, when the speed increases, *W* moves radially and *F*

moves towards the center of the shaft, allowing A to go to the right and B to drop into the notch N, **when the exhaust valve is open.** This keeps the exhaust valve open during the succeeding suction stroke so that no charge is drawn in. B remains in N, keeping the exhaust valve open, until the speed lowers and F forces A to the left once more. Even then B cannot escape from N until the exhaust cam gets into the open position and relieves B of the tension of spring S_3.

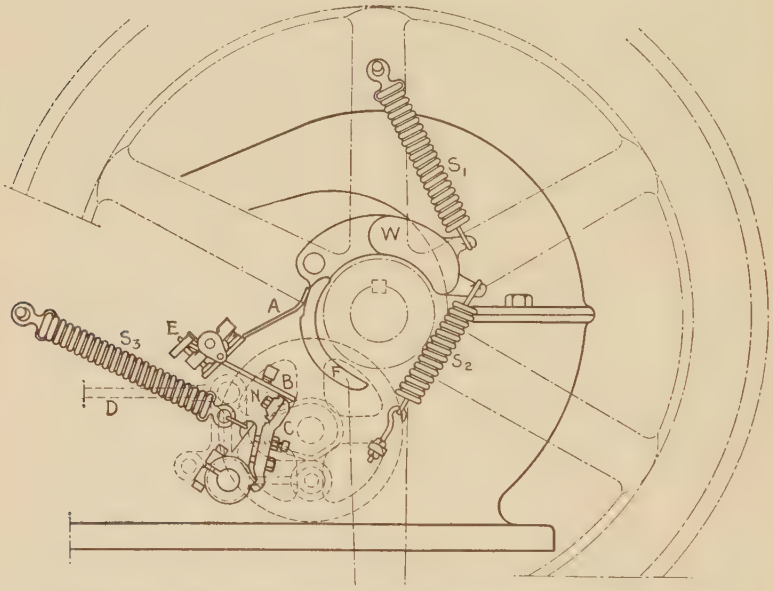


Fig. 54. — Centrifugal Mechanism for Hit-and-miss Governor. Seager Gas-engine Works.

Fig. 55 illustrates a type of inertia governor built by Crossly Bros., Ltd., Manchester, England, and used on their small four-cycle engines. The cam A actuates the roller B at the proper time for opening the inlet valve. The governor arm G carries on one end the weight E and on the other the pick-blade H. The valve-rocker D, which transmits the motion from the roller B to the valve-stem L, is fulcrumed at C, and carries the governor arm hinged to it at F. The governor arm, with the weight and pick-blade, is thus free to swing through a small arc relatively to the rocker D, except that the spring S_3 holds it against the rocker and

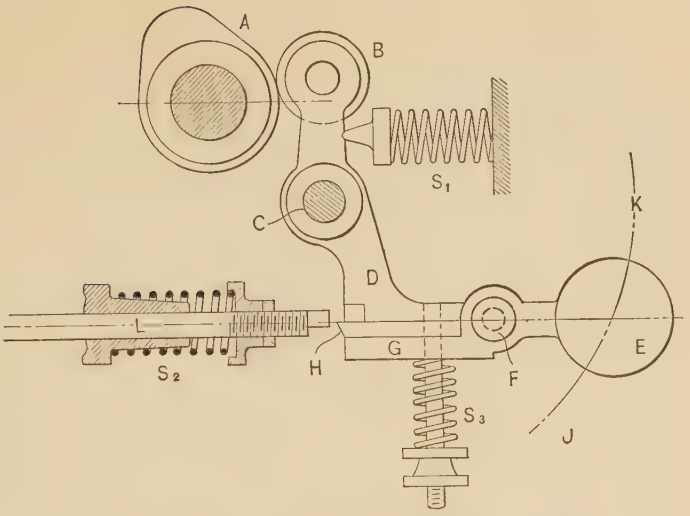


FIG. 55. — Inertia Type of Hit-and-miss Governor.

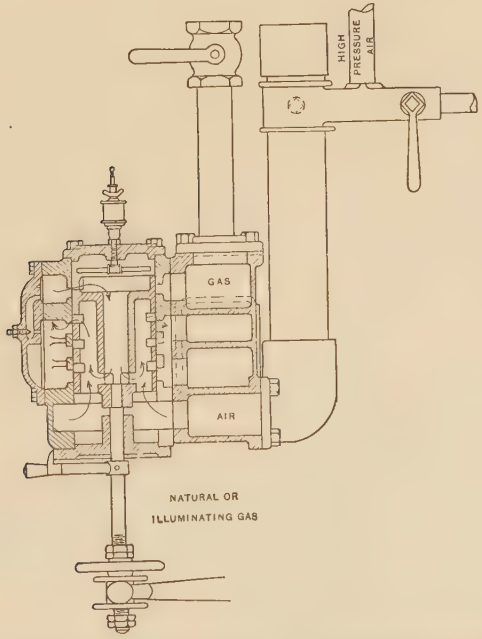


FIG. 56. — Mixing and Throttling Valve. Bruce-Macbeth Engine Co.

permits the pick-blade, under normal conditions, to meet the end of the valve stem L. S_2 is the main valve spring and S_1 holds the roller B to the cam.

Under normal operation the weight E moves in the arc JK. However, when the speed increases, the inertia of the weight E causes it to lag behind so that it revolves about the center F. H then does not come in contact with the end of L, and the inlet-valve remains closed for that stroke.

Throttling Governor. — In Fig. 56 is illustrated the mixing and throttling valve used by the Bruce-Macbeth Engine Company of Cleveland, Ohio. Gas enters from the pipe marked "gas" through the port in front, as shown by the arrow, and passes down through the center of the piston valve. At the bottom of this valve it mixes with the incoming air at practically atmospheric pressure. The mixture then passes up the annular space in the valve and out through the three ports to the inlet valve chamber between the passages marked "gas" and "air" respectively.

The piston valve has no rings but is ground to a fit. The only adjustment is at the handwheel shown, which changes the proportion of gas to air. The three ports shown do not open and close together; in fact, the top port is cut on a $\frac{3}{16}$-inch spiral. At a point where this spiral is fully opened, the second port begins to open and after this second port is open $\frac{1}{32}$ inch, a third port begins to open.

The needle-point screw in the cover of the valve chest allows gas to be by-passed from the gas-pipe directly into the inlet valve chamber, around the throttle valve. This hole is but $\frac{3}{16}$ inch in diameter for a 150-horse-power engine, but a slight adjustment of the screw gives the proper mixture and consequently maximum efficiency.

Variable Lift. — Throttling is sometimes accomplished by means of a variable lift on the inlet valve. This is usually done by using a movable fulcrum in connection with a valve rocker arm. This arrangement is illustrated in the Deutz valve gear, Fig. 57. A shows the position at full load and B at light load. The gas passage is above the air passage, and is closed by a separate valve on the main stem. The main valve controls the air admission. As the valve lifts, a rush of air enters the cylinder followed by a mixture of gas and air. If there is a slight play between the shoulder of the gas valve and the sleeve above it, the

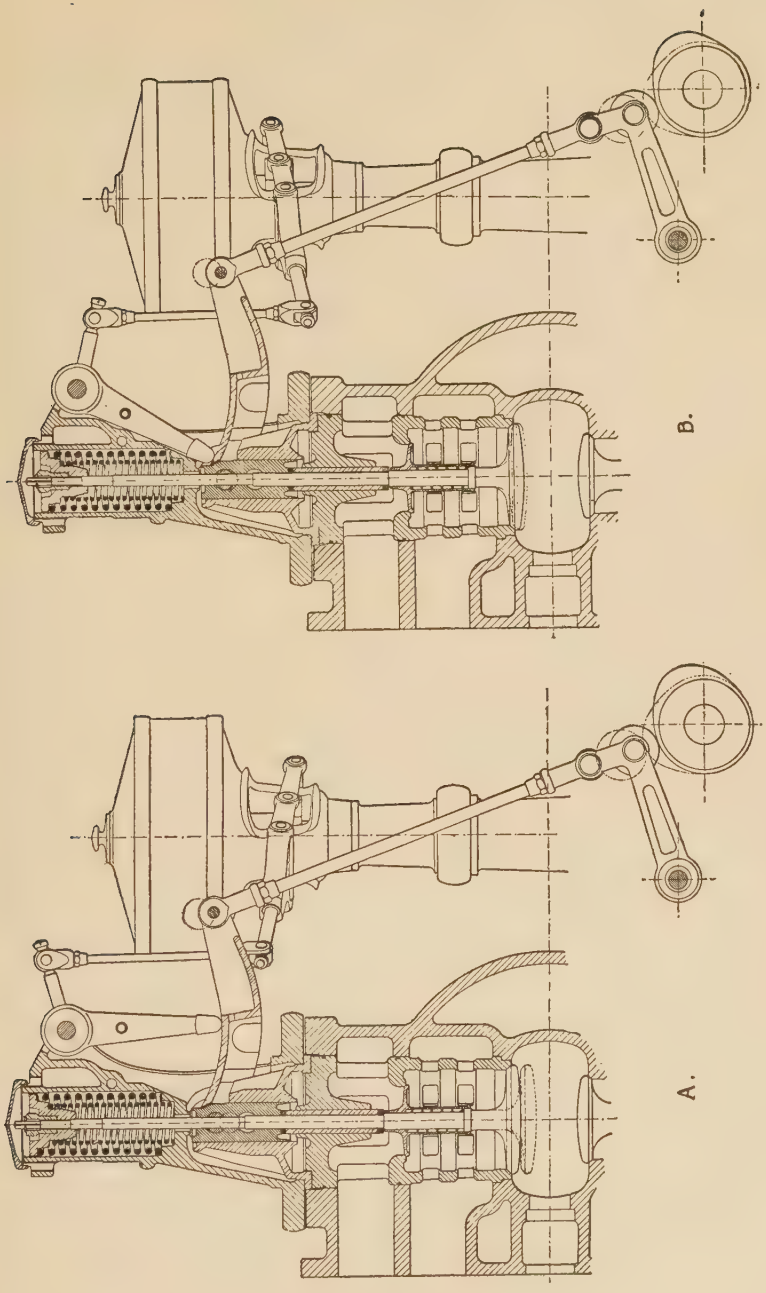


FIG. 57.—Deutz Valve Gear and Governor.

main valve will open slightly ahead of the gas valve but the gas-valve will close before the main valve. It will be noticed that there is a small spring within the gas valve sleeve which keeps this valve against the seat until forced off by the sleeve on the main stem.

This gear is practically a constant-quality device except at very light loads. The variation in quantity, and hence in the compression, is clearly shown in Fig. 58, which illustrates cards taken from a Deutz engine at full, half and light loads.

The valve and governor mechanism of the Olds engine, now built by the Seager Gas-engine Works, is illustrated in Fig. 59. In this engine there is a combined gas and inlet valve on the same stem, the inlet valve being of the ordinary mushroom type, while

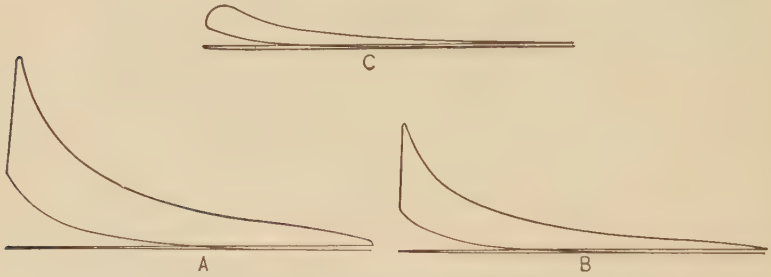


Fig. 58.—Indicator Cards from Engine Equipped with Deutz Valve Gear and Governor.

the gas-valve is the sleeve K above the inlet-valve. This gas valve has a lap so that the main valve opens appreciably before the gas is admitted. The fulcrum block H is moved out and in by the governor. At low speed full load the block is to the right, giving maximum opening to the valves. As the speed increases the block is moved to the left and the lift of the valve is decreased. As the lift decreases, the ratio of gas to air becomes smaller, or the mixture becomes leaner. The quantity decreases, also, so that at light loads the compression is much lower than at larger loads.

Constant-quantity Governor.—In constant-quality governing, whether it is done by actual throttling, by cut-off, or by changing the lift of the valve, compression at light loads decreases very appreciably, enough so that the ignition is uncertain, and often

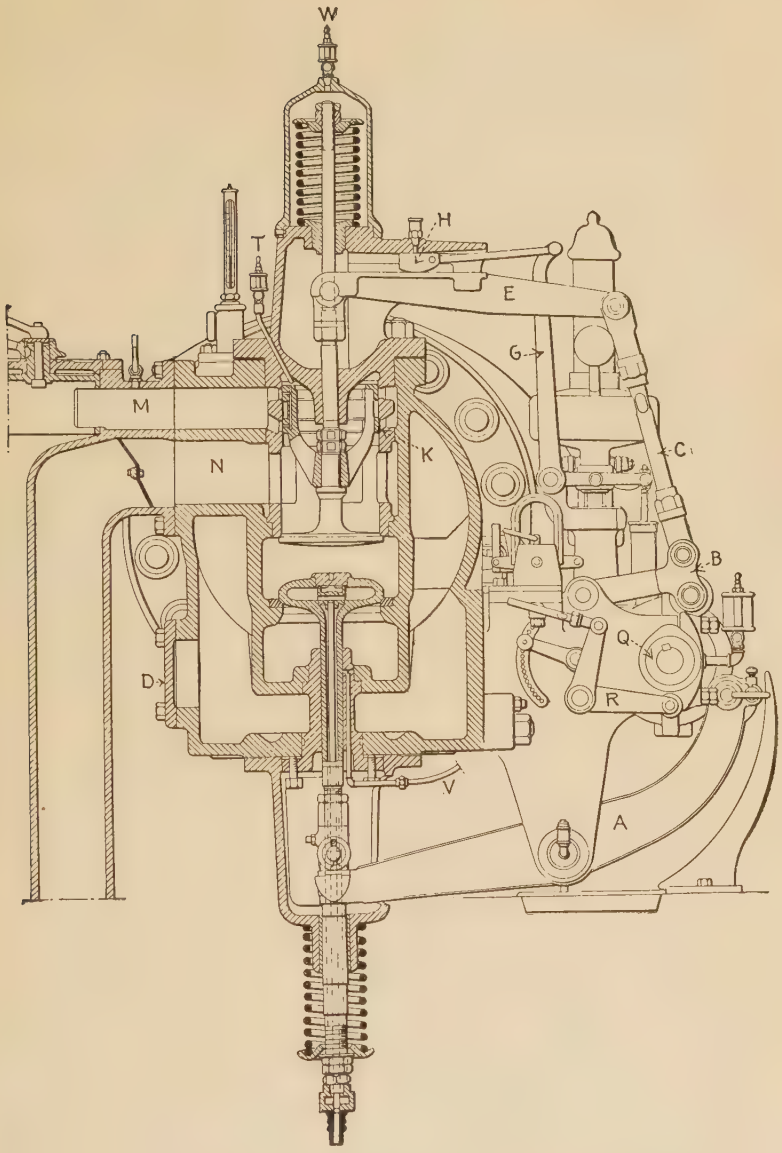


FIG. 59.—Valve and Governor Mechanism. Seager Gas-engine Works.

fails entirely for several strokes in succession. The author has made tests on engines which apparently indicated higher power at light loads than at heavier loads. The reason for this was that on light loads the ignition would fail entirely for a stroke or two

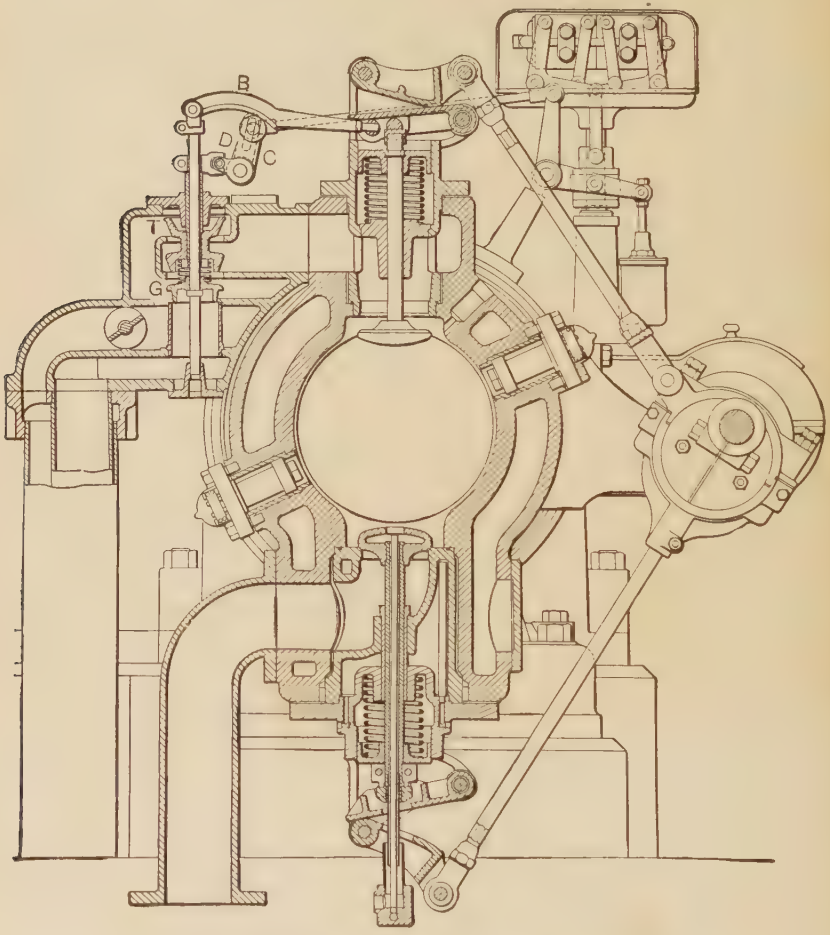


Fig. 60.—Buckeye Valve and Governor Mechanism.

and the engine would slow down enough to get a charge of good gas, then an explosion would occur that would give a higher mean effective pressure than the average. In order to get the proper indicated power it was necessary to count the missed explosions as on a hit-and-miss engine.

In order to overcome this difficulty of missing explosions at light loads, several forms of governors have been designed. One of these is the one used on the Buckeye engine and is illustrated in Fig. 60. The exhaust valve at the bottom and the inlet valve at the top of the cylinder are operated by the same eccentric through rolling levers. Each combustion chamber is supplied with fuel by a separate mixing chamber. The gas enters the lower compartment of each mixing chamber through the small pipe enclosed in the air pipe. A gas valve *G* prevents the gas and air from mixing at all times except during the suction stroke. Valve *G* derives its motion from the main inlet valve lifter through the segment lever *B* which fulcrums on a roller carried by lever *C*. This lever is under the control of the governor with the result that while the gas valve lifts every suction stroke, the extent of the lift is variable, being greater at heavy loads and less on light loads. Surrounding the gas valve stem is a tubular stem attached to the balanced throttle valve *T*. This valve is also governor-controlled through the lever *D*. It should be noted, however, that while the gas valve opens and closes every suction stroke, the lift varying with the load, the throttle valve is held stationary as long as the load is constant and moves to a new position only when the load varies. At all loads above 60 per cent of the rating, this valve is held so high above its seat that the mixture is not throttled appreciably, the governing being done entirely by varying the lift of the gas valve. As the load falls off, the increased speed causes the governor weights to move the fulcrum roller under the segment lever to a position where the gas valve will have a decreased lift; and at the same time the action of the governor reduces the opening of the throttle valve. All loads from maximum down to about 60 per cent are handled purely by varying the strength of the mixture, while the total volume of the mixture and, consequently, the amount of compression remain unchanged during this part of the load range. At about two-thirds load the throttle valve begins to reduce the volume of mixture admitted to the cylinder so that all lighter loads are carried by weakening the charge and using less of it.

Since lean fuel mixtures burn more slowly than rich ones it has been found advisable to vary the ignition timing as the load on the engine varies. The governor does this automatically by advancing the point of ignition on light loads and retarding it on heavy ones.

Advantages of Different Governing Systems.—For small engines driving machinery which does not require particularly close regulation, the **hit-and-miss** system is to be preferred. It is cheap and simple to build, and, being simple, does not get out of order easily. On a four-cycle single-cylinder engine one missed explosion means that the engine must make eight strokes or four revolutions with one power stroke, which shows conclusively that close regulation cannot be expected.

The plain throttling governor is next to the hit-and-miss as regards simplicity and low cost. If the throttle valve admits the proper amount of correct mixture at full load to give maxi-

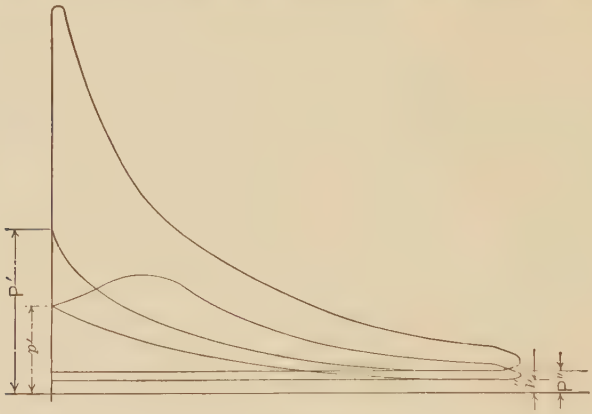


Fig. 61.

mum efficiency, at loads less than the full load, the amount admitted will be less and efficiency will drop off rapidly. Fig. 61 shows two cards from a throttle-governed engine, one at full and one at a lighter load. The one at full load has a pressure at the beginning of compression of 14 pounds, P'', and a pressure at the end of compression of 125 pounds, P'. The light-load card has pressures at these points of 10 and 95 pounds, respectively. The ratio of full mixture, by weight, at beginning of compression is 1.4. The mean effective pressures of the two cards are 60 and 20 respectively, or the large card shows 3.0 times the work of the small card on 1.4 times the fuel. Hence the efficiency of the large card is $\frac{1}{1.4} \times 3.0 = 2.14$ times as great on the full load as on the light load.

The loss in efficiency here is due to filling the cylinder only to 10 pounds absolute pressure at the beginning of compression instead of to 14 pounds as on the full load. It will be noticed that a difference of 4 pounds at the beginning of compression makes a

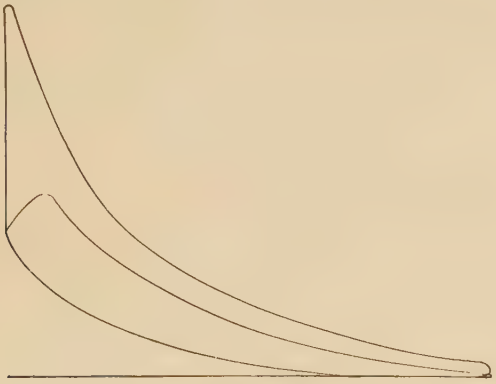


Fig. 62.

difference of 30 pounds at the end of compression. One way to avoid part of this loss is to dilute the mixture at light loads with enough air to increase the compression up to the normal.

Fig. 62 illustrates two cards showing the effect of diluting the

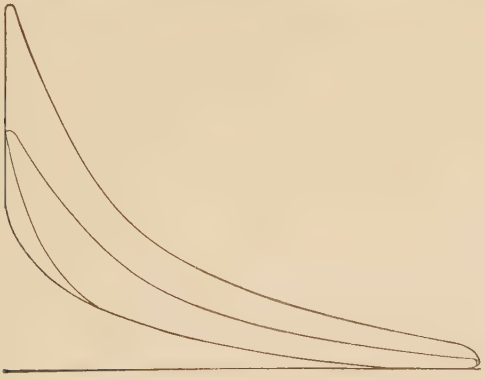


Fig. 63.

mixture at light loads to bring up the compression pressure. Little is gained here as the increased dilution practically offsets the advantage gained by increased compression, and slow burning is the result as before.

The method employed in the governor shown in Fig. 60 is to advance the spark and start combustion earlier at light loads. Since the weaker mixture is slower burning than the normal mixture, early ignition is the remedy for the late burning shown in the cards in Figs. 61 and 62. The effect of advancing the spark and making ignition earlier is shown in Fig. 63.

Still another method which has been tried, but has not been found entirely satisfactory, has been to use more of the mixture

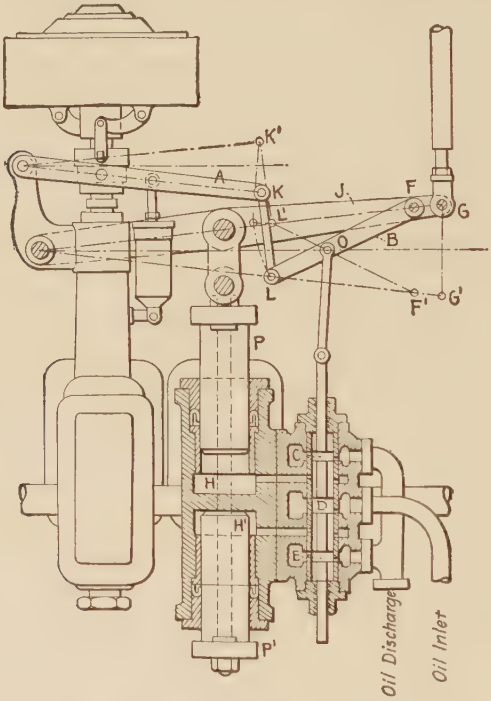


FIG. 64.—Power Governor Used on Large Gas Engines.

at light loads than at full load, to insure higher compression at the light loads. The only way to do this is to cut down the amount of mixture at full loads by throttling, thus reducing the pressure at which compression begins. If, by this method, proper compression and therefore maximum efficiency is secured at light loads, the compression at full loads will be below that required to give best efficiency. Thus, while efficiency is increased at one point, at a more important point it is decreased.

Power Governors.—In large gas engines, particularly the twin-tandem type, the power of the centrifugal governor is insufficient to operate the valves and other means must be resorted to. In such cases the governor proper usually operates a small pilot valve which in turn admits oil under pressure to plungers which move the governing valves. In Fig. 64 the governor operates the pilot valve *CDE*. The plungers *P*, *P'*, actuate the lever *J*, which regulates the engine valve gear. The governor is shown in its low position, the governor lever *A* is down and the hydraulic plungers are held up by the pressure of oil which has been admitted above *D* and into *H*. If speed increases and the governor rises, it will lift the pilot valve. The oil under pressure will then be admitted below *D* and into *H'*. At the same time the oil in *H* will be discharged through the upper port which is uncovered by spool *C*. The hydraulic plungers will thus descend, bringing down the lever *J*, and the pin *F* moves towards the position *F'*, until the pilot valve again covers the three ports and stops all motion of the governor. This position is held until another change of speed occurs, when the pilot valve will be moved up or down as the speed increases or decreases, the plungers being moved in a corresponding direction. Thus at any given speed, the governor has a fixed position.

CHAPTER XI

COOLING ENGINES

Necessity for Cooling — Cooling Systems

General. — It is necessary to furnish some means of cooling at least the cylinders of all internal combustion engines. The amount of heat generated in the cylinder of such an engine varies from 7500 to 15,000 B.t.u. for each horse-power hour developed. From 25 to 35 per cent of this heat must be conducted away and, within certain limits, the lower the temperature at which conduction takes place the better will be the conditions for the engine.

The amount of heat that will flow per unit of surface from a gas engine cylinder depends on first, the time; second, the difference in temperature; third, the material of cylinder wall; fourth, the thickness of cylinder wall; fifth, the condition of cylinder wall on the inside and outside. We may select any unit of time for the discussion and since the hour is the most convenient, there need nothing more be said in regard to the time. To illustrate the other items, we may refer to the sketch in Fig. 65. The heat must first flow through the surface *A* into the wall of the cylinder *B*. This cylinder wall is usually of cast-iron and of a thickness varying with the diameter of the cylinder and the pressure. The condition of the cylinder wall inside is fixed by the necessity of lubrication, so that the only condition which is really variable is the outside surface of the wall. If the wall *C* is left smooth, or, in other words, if *C* and *A* are of the same area, and the air surrounding *C* is still, the heat that will flow per hour from *A* to *B* and from *B* to *C* would not take care of the excess that must be radiated, unless the temperature difference between *C* and the air is very great, so great in fact that the metal becomes injured.

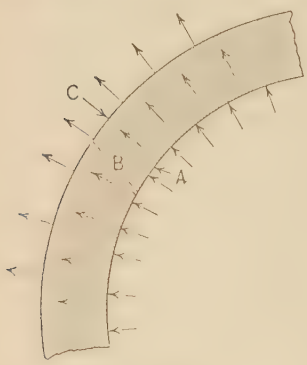


Fig. 65.

If we change the character of the surface C by roughing it up and increasing its area, keeping the area of surface A constant, we have a condition shown in Fig. 66. With this condition more heat would flow from surface C than from surface A, since for one square foot of A we might have several square feet of C. But since the heat is constant, that is, since as much flows from C as into A, another condition must change. This condition is the temperature of the surface C. Since there is more outlet, as it were, for the heat at C, the temperature of B will drop lower in Fig. 66 than in Fig. 65. To further increase the flow of heat from C to the air, the air surrounding C is caused to move. This is usually done by means of a fan.

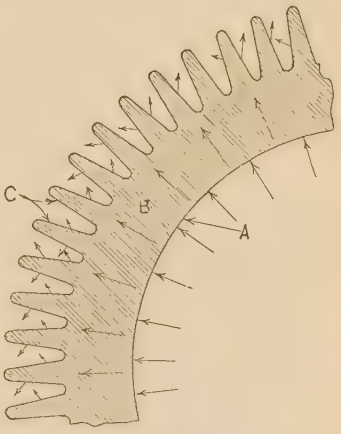


FIG. 66.

Air-cooled Engines.— Small engines may be cooled successfully by increasing the area of the outside surface of the cylinder as illustrated in Fig. 66. The area is increased in many ways. One of the first air-cooled automobile engines built in the United States was made with numerous copper spines, the size of a lead pencil, screwed into the cylinder wall, until the cylinder presented a grotesque appearance, not unlike a mechanical porcupine. Another method is used to cast deep, thin ribs on the cylinders. In some cases these ribs extend around the cylinder transversely as in the motor cycle engine, while in others the ribs are longitudinal, parallel to the axis of the cylinder.

In Fig. 67 is shown an air-cooled engine, built by the "New Way" Motor Company of Lansing, Michigan. The cylinder has cooling fins arranged so that they are perpendicular to the axis of the cylinder. These fins are perforated with holes from $\frac{1}{4}$ to $\frac{3}{8}$ inch in diameter. The fins are also connected by bars or rods running at right angles to them. A cooling fan in the flywheel forces air through these fins, and the bars and the holes break up the air currents so that the air is heated uniformly and no stream lines can exist. The cylinder is surrounded by a sheet-metal jacket which fits closely against the fins, so that the

air that is forced in by the fan must pass between the fins and through the holes.

Experiments have shown that the best results are obtained on this engine when a thermometer inserted between two fins on the cylinder head registers between 300 and 350° F. When one of the engines was operated on a 14-day full-load test, the temperature did not drop below 310 degrees, nor rise above 340 degrees, as shown by half-hourly readings.

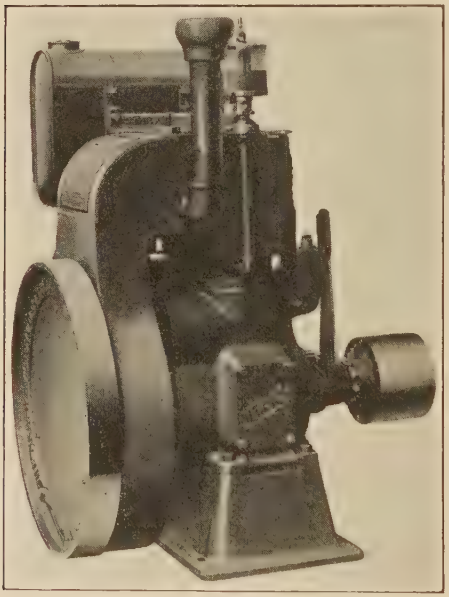


FIG. 67.—"New Way" Air-cooled Engine.

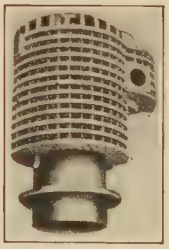


FIG. 68.—Cylinder of "New Way" Engine.

The Franklin automobile motor shown in Fig. 90 is cooled by air. Longitudinal fins made of sheet steel are cast into the cylinder walls. These fins are encased by a jacket of steel, open at the top and bottom. A fan of the Sirocco type placed in front of the engine forces air down between the fins and the surrounding jacket. The hood that covers the engine is arranged in such a way that there is no path left for the air except through the path inside the jackets.

Water Cooling.—Some manufacturers of small engines, and all manufacturers of large engines, cool the engine cylinders

with water. When water is used for cooling, the cylinder is made with a double wall, the space between the two walls being the **water jacket.**

Single-acting engines are usually cooled only around the barrel of the cylinder and on the cylinder head. When the diameter of the cylinder is above 20 inches, however, the piston must be

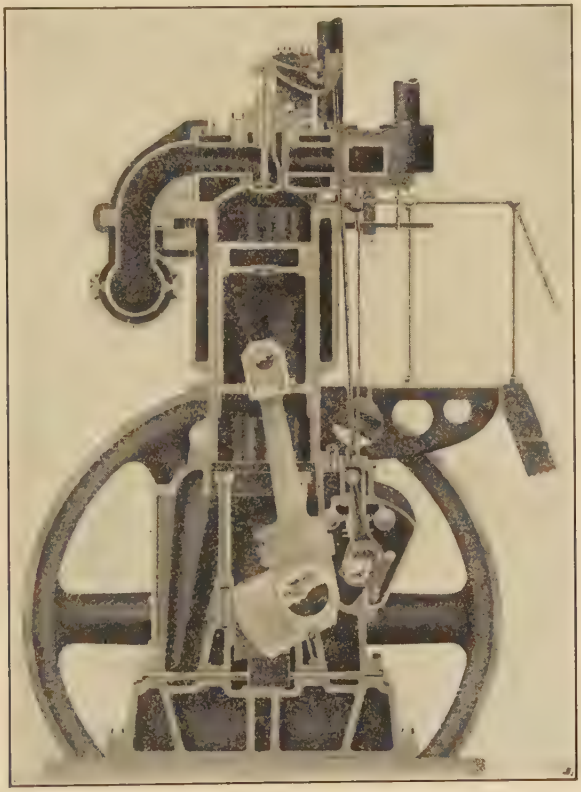


FIG. 69.—Water-cooled Vertical Gas Engine.

cooled. In double-acting engines, both the piston and rods must be water cooled, as there is no chance for radiation as there is in a single-acting engine.

Water cooling is more effective than air cooling for this reason: The heat conducted away from a surface surrounded by water amounts to about 100 times more than that conducted away from a similar surface surrounded by air. Of course this ratio depends

entirely on the motion of air and water. Much more heat is conducted away by air in motion than by still air.

Fig. 69 shows plainly how the water jackets are arranged in a vertical engine. It will be noticed that the inner and outer cylinder walls are cast in one piece, and that the jacket extends down about to the location of the top of the piston in its lowest position. It is also evident that the cylinder head and the exhaust manifold are water jacketed.

In larger engines the outside wall of the cylinder is cast separate from the inside wall. This is done to prevent shrinkage strains in the castings and complicated core work. Fig. 70 shows one method of inserting this **liner,** as it is called. The liner is bolted rigidly to the outside wall at the upper end. A copper gasket inserted at *A* prevents leakage at that point. At the lower end of the cylinder a stuffing-box may be formed. This makes a water-tight joint and allows for expansion of the liner.

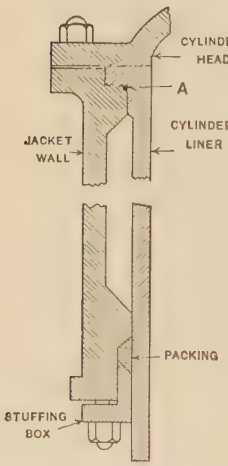


FIG. 70.

Pistons and Rods.—As stated above the pistons and piston-rods of double-acting engines must be water cooled. To accomplish this the rods and pistons are made hollow as shown in Fig. 71, which illustrates the arrangement of rods, etc., in the tandem double-acting engines built by the C. & G. Cooper Company of Mt. Vernon, Ohio. Water enters at the intermediate crosshead and flows in both directions, that is, toward the main crosshead, through the front piston, and toward the rear crosshead, through the back piston. The water enters the pistons at the bottom and on becoming heated rises and leaves by the top. At the front and rear crossheads the water escapes into long, trough-shaped funnels.

The water inlet at the intermediate crosshead gives trouble if not constructed with care. The best type appears to be a telescopic pipe, the inner or moving pipe being of bronze. A generous stuffing-box must be provided and air chambers used to prevent surges of water due to the pumping action. The water pressure for such an engine should be at least 40 pounds, and where water

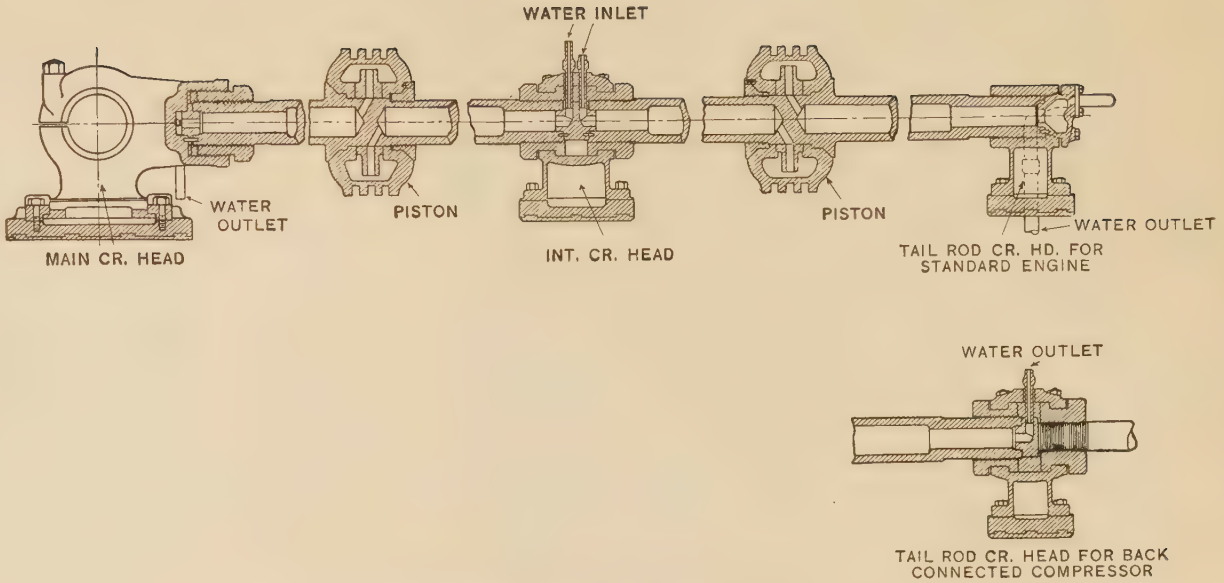


FIG. 71.—Water-cooled Pistons and Rods. C. and G. Cooper Co.

at that pressure is not available, a triplex pump, driven from the main engine, may be used to drive the water through the cooling system.

Cooling the Water.—In stationary engines no attempt is made to cool the water and use it over again except where the cost of the water makes the installation of a cooling system a necessity. For large engines cooling towers, similar to those used in connection with condensers, may be and are used. Some makers of small engines carry out the same idea by using conical towers made of galvanized iron. For automobile work the well-known

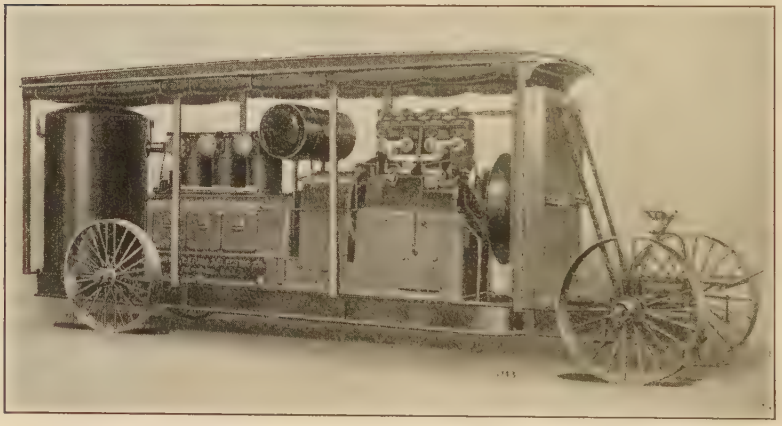


Fig. 72.—Portable Gasoline Engine Driving Air Compressor, Jacket Water Cooled in Radiator.

cellular or tubular radiator is used. These are also sometimes used with portable engines as shown in Fig. 72, which illustrates a portable three-cylinder, vertical air compressor driven by a four-cylinder vertical gasoline engine. The large rectangular radiator is shown at the front of the machine. Fig. 73 shows a vertical hopper-cooled engine, a type popular on construction and other transient work.

Water Required.—A very simple calculation will suffice to determine the amount of water required to cool a gas engine. For an extreme case let us assume 15,000 B.t.u. generated in the cylinder per horse-power hour, and that 35 per cent of this heat

is absorbed by the jacket water. The amount of heat to be absorbed per horse-power hour will then be

$$0.35 \times 15{,}000 = 5250 \text{ B.t.u.}$$

We may assume that the water enters the jacket at 70° F. and leaves at 190 degrees. The rise in temperature will then be 120 degrees. The water per horse-power hour will then be

$$\frac{5250}{120} = 43.7 \text{ pounds.}$$

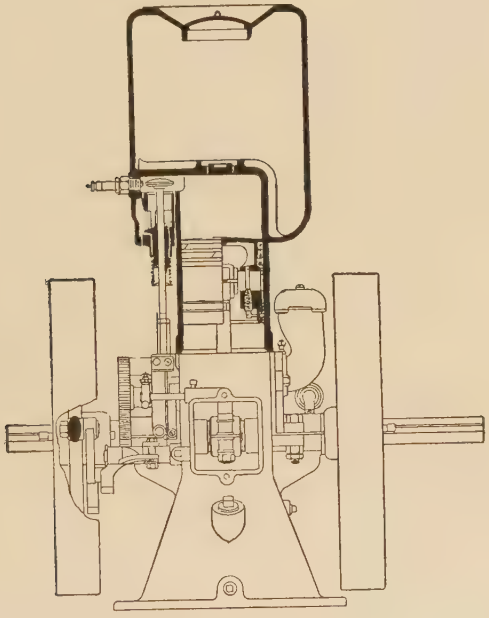


FIG. 73.—Vertical Hopper-cooled Engine. Novo Engine Co.

Recapitulating, we have

$$W = \frac{xH}{100(t_1 - t_2)},$$

where

W = weight of water per horse-power hour in pounds,
H = B.t.u. supplied to engine per hour, per horse-power,
x = percentage of heat absorbed by jacket,
t_1 = temperature of water leaving jacket, degrees Fahrenheit,
t_2 = temperature of water entering jacket, degrees Fahrenheit.

If the water is not allowed to leave the jacket of the engine but is held there until it **evaporates,** much less water is needed as each pound of water or steam carries away with it the latent heat of vaporization in addition to the sensible heat. The formula for the amount of water necessary per horse-power hour becomes

$$W = \frac{xH}{100[(t_1 - t_2) + r]},$$

where r is the heat of evaporation at atmospheric pressure, approximately 970. With the conditions given in the previous problem, that is, with 15,000 B.t.u. generated in the cylinder per horse-power hour, 35 per cent of which is absorbed by the jacket water, the water used per horse-power hour will be

$$W = \frac{xH}{100[(t_1 - t_2) + r]} = \frac{35 \times 15{,}000}{100(120 + 970)} = 4.8 \text{ pounds.}$$

CHAPTER XII

TYPES OF ENGINES

Small Gas and Gasoline Engines

General.—There are so many ways in which different types of internal combustion engines may be classified that it would be out of place for anyone to classify them in one way, and claim that way to be the best. In this and the two succeeding chapters the author wishes to illustrate and describe briefly some of the types of engines made in the United States and abroad. In order to avoid confusion a partial classification has been attempted. It is as follows:

1. Small gas and gasoline engines.
2. Automobile engines.
3. Large gas engines.
4. Oil engines.

There is no necessity of differentiating between gas and gasoline engines. With the exception of the carburetor and the mixing valve there is no inherent difference. In fact, the carburetor may be taken off a gasoline engine and a mixing valve substituted, and the engine will work efficiently on any rich gas. Hence, the gasoline and gas engines are put under one classification.

The engines that are shown in the following chapters have not been selected for their superiority; they represent types of engines which are generally conceded to be good practice today, and which show the trend of the development for tomorrow.

SINGLE-CYLINDER ENGINES

Foos Single-cylinder Horizontal Type.—The engine illustrated in Fig. 74 is built in sizes from 8 to 60 horse-power. It is of the four-cycle type, built to operate on gas and liquid fuels, and is water cooled. The valves are attached to the sides of the cylinder and not to the head. They are both vertical and both mechanically operated from the bottom. Both valves are in

separate detachable chambers surrounded by water-cooling jackets.

The jacket wall is cast solid with the cylinder. The water space is closed by the cylinder head and when the latter is removed an opening is exposed sufficiently large for the removal of any accumulation.

The piston is of the trunk type, made of close-grained cast-iron, and carries four rings.

Ignition is of the make-and-break system using an oscillating magneto for producing the current. The gear shown between

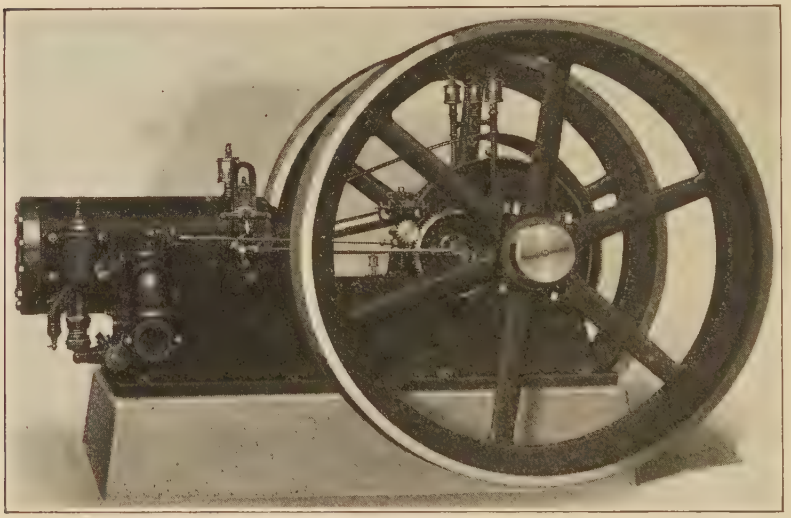


Fig. 74.—Foos Single-cylinder Engine.

the spokes of the flywheel in Fig. 74 operates the igniter rod which is seen extending up to the magneto. The cams which operate the valves are on the same shaft with this gear, between the gear and the crank discs.

The governor is of the centrifugal, fly-ball type and operates a fuel valve of the double-poppet type. This valve is balanced regardless of its position or the amount of suction due to the variation in the load.

The shaft is of the center crank type and is a solid forging. The balancing weights are in the form of discs and fitted to the machined crank-arms as shown in Fig. 75.

The connecting-rod is of the marine type at both ends, is of circular section and is fitted with phosphor bronze-bearings.

The use of small electric generating sets for farms, country homes, isolated churches and similar buildings has become very broad. A complete unit except batteries, manufactured by the Delco-Light Co., Dayton, Ohio, for this class of service is shown in Fig. 76. Engines for this service are carefully designed for simplicity and compactness. They are operated without expert attention, very frequently in places where there are no adequate facilities for repairs. A careful study of the illustration will well repay the reader.

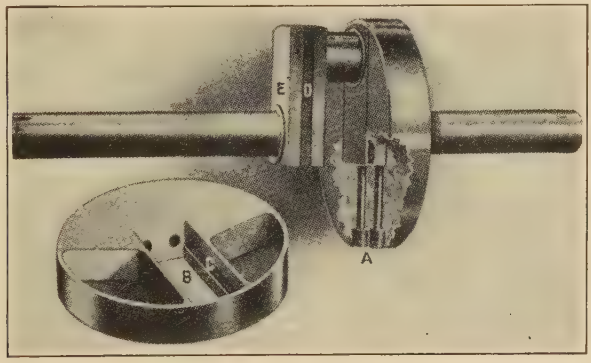


Fig. 75.—Crank-shaft of Foos Horizontal Engine.

The unit is air-cooled. The cylinder has generous fins cast on it and is surrounded by a metal jacket. The flywheel is cast in the form of a multi-vane fan which draws air down inside the jacket around the fins and exhausts it around the periphery.

The piston is made of aluminum and carries three rings.

The crank-shaft is counter-balanced and is carried on one roller and one ball-bearing. The generator has no outboard bearing.

The valves are in the cylinder head and are both operated by vertical push rods from the cam-shaft which is in the crank-case.

Lubrication is accomplished entirely by the splash system. The narrow-faced gear shown below the crank-shaft dips into the oil in the engine base and throws it into all the bearings and on the cylinder walls.

This engine, which is built to operate on gasoline, kerosene, distillate or natural gas uses a mixing valve instead of a carburetor.

The generator acts as a starting motor until the engine starts to operate under its own power. The generating of current then begins automatically. The current may be used direct from the generator for lights, motors, etc., or may be used to charge storage

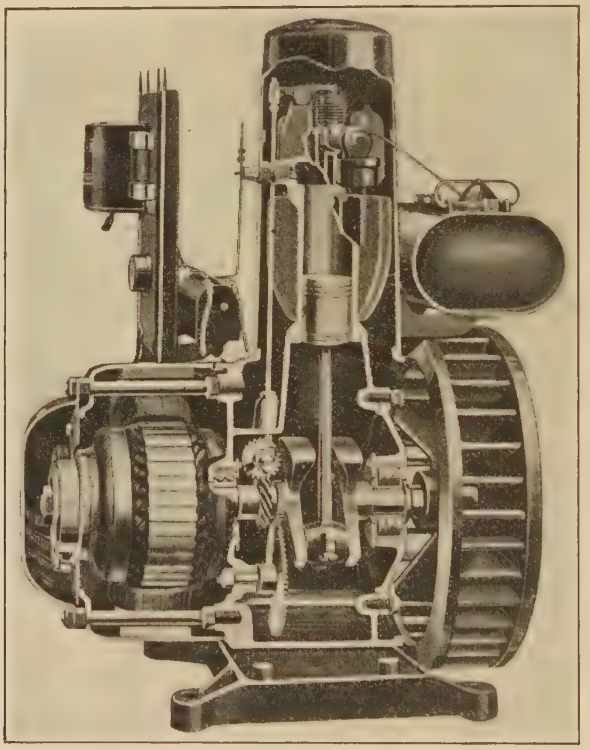


Fig. 76.—Section through Delco-light Unit.

batteries. If used for the latter purpose, the unit stops automatically when the batteries are fully charged.

These units are built in capacities of 850 watts, $1\frac{1}{4}$ kilowatt and $2\frac{1}{2}$ kilowatt. The first two operate at 32 volts and the last at 10 volts.

The Otto Engine.—The Otto engine shown in Fig. 77 is made in single-cylinder units in eight sizes from 40 to 170 horse-power,

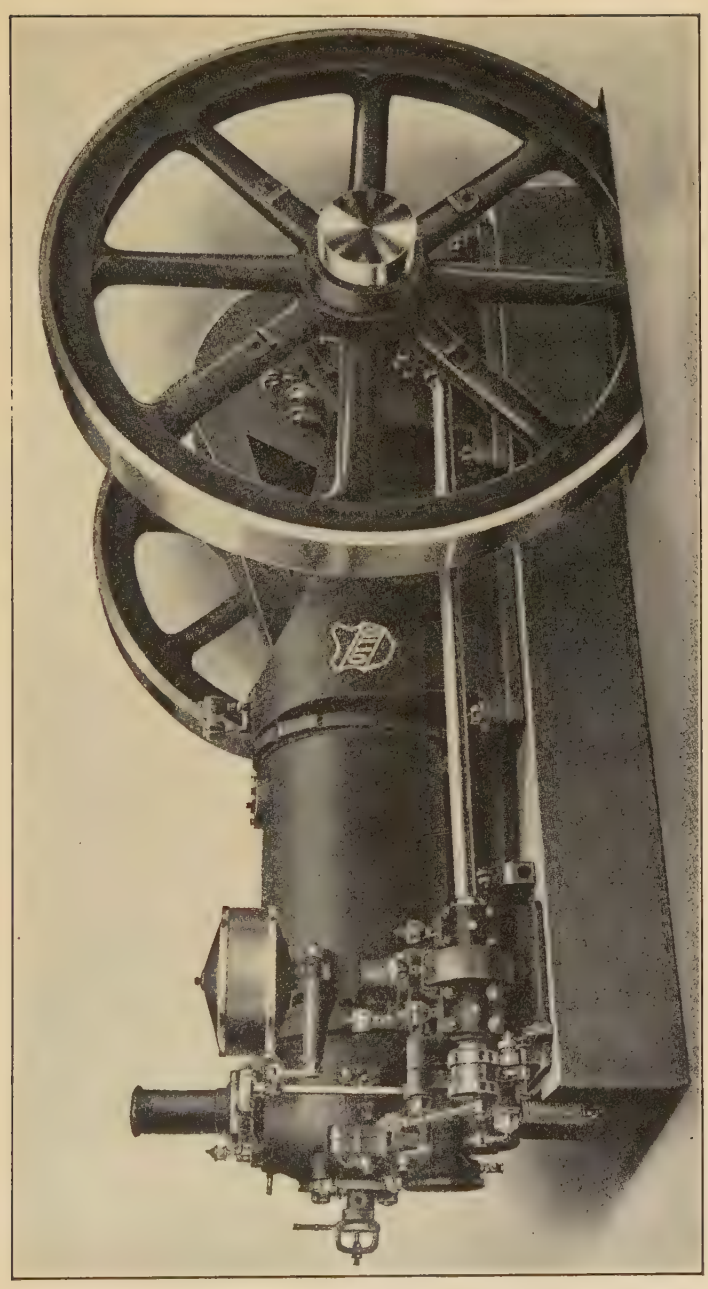


FIG. 77.—Otto Single-cylinder Gas Engine.

and in twin-cylinder units in eight sizes from 80 to 340 horsepower. The engine works on the four-stroke cycle.

The **frame** and **cylinder** are cast separate, and the frame extends forward so as to serve as a base for the cylinder and head. These parts simply rest on this base and are not fastened to it, thus permitting free expansion and contraction of the heated parts.

The **cylinder** and **jacket** are cast in one piece, with large water spaces. Separate cooling-water connections are provided for cylinder and head, so that water to these parts may be adjusted separately. A solid wall separates the cooling-water spaces in the cylinder and head, so that no cooling water passes through a packed joint.

The **piston** is of the trunk type and is made of the same grade cast-iron as the cylinder. The pin is held in place by set-screws and keys.

The **cylinder head** contains the combustion chamber. All valves, together with the igniter, are mounted on it. The entire head is water jacketed. The valve seats are removable and may be easily replaced in case of wear. The valve stems are provided with renewable bushings.

The **crank-shaft** is of 0.40 carbon open-hearth steel, forged from the billet and polished all over. The counterbalance weights are fastened to the crank-arms.

On single-cylinder commercial engines two **flywheels** are provided. On the direct-connected special electric type and on the twin-cylinder type, but one extra heavy flywheel is used.

The **connecting-rod** is of 0.20 carbon open-hearth steel. The boxes at each end are of the marine type.

Ignition is accomplished by means of an oscillating magneto, mounted on the cylinder head and positively driven. The movement of the armature in this magneto is limited to a few degrees. No spark coil is necessary. The igniter is of the make-and-break type, consisting of a flange in which are housed the fixed and movable electrodes. An interrupter lever and spring complete the device. The igniter lever, operated by a crank disc on the gear shaft, oscillates the armature of the magneto and separates the electrodes at the proper time, causing a break in the circuit, thus producing the spark.

The **gear shaft** is driven from the crank-shaft by spiral gears running in oil. This shaft carries the spiral gears that drive the

governor. On it are also mounted the cams that actuate the inlet and exhaust valves. The gear shaft is carried in bearings fastened to the frame at both ends, permitting the removal of the valves, cylinder head and cylinder without dismantling the gear shaft, nor is the alignment of the gear-shaft bearings affected

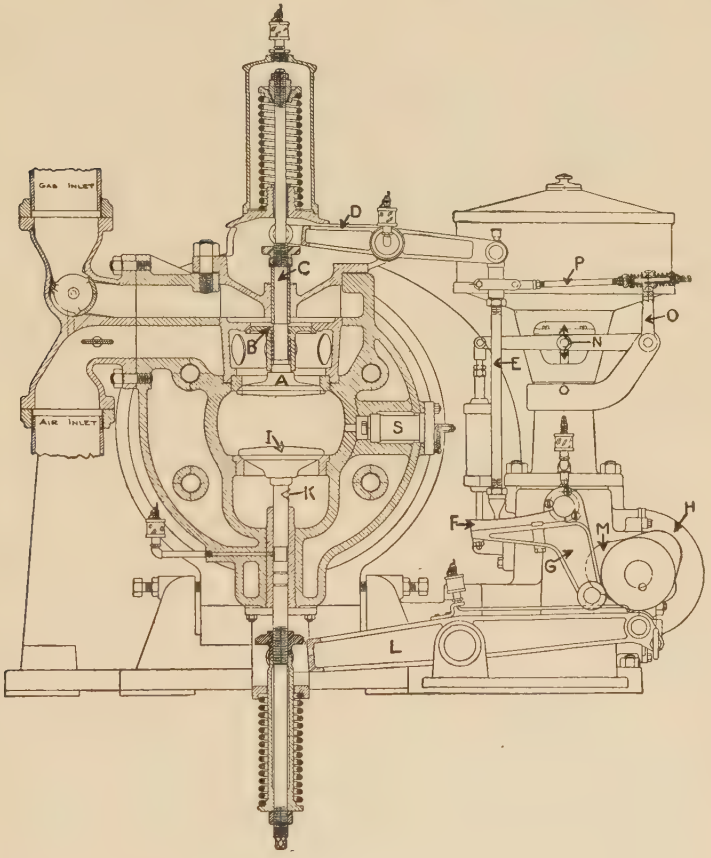


FIG. 78.—Otto Gas Engine. Section through Combustion Chamber.

by the expansion and contraction of the cylinder head or cylinder jacket.

Governing is accomplished by means of a high-speed ball-type governor which controls the lift of the mechanically operated inlet-valve, regulating the amount of the mixture of air and gas according to the load. In Fig. 78 is shown a section through

the valves and combustion chamber. It will be noticed that the charge enters the cylinder through two separate ports, controlled by hand-operated throttles and the poppet inlet-valve *A*. The poppet valve *B*, mounted flexibly on the valve-stem *C*, times the admission of the gas in relation to the air inlet. The inlet-cam *H*, acting upon the lever *G*, upon which the steel shoe *F* is mounted, opens the inlet valve *A* in a positive manner by means of the push rod *E* and the valve lever *D*. The lift of the valve *A* is decreased or increased when the pivot pin *N* is raised or lowered by the governor, thus causing the lower end of the push rod *E* to be moved by means of lever *O* and rod *P* close to or farther away from the fulcrum of the lever *G*. There is always a slight amount of play between the end of the rod *E* and the steel shoe *F*, so that the work performed by the governor in swinging the rod *E* is very light. A dash-pot consisting of a cylinder and piston, oil cushioned and connected to the governor lever, steadies the operation of the governor and prevents sudden variation of speed under fluctuating loads.

The **exhaust valve** *I* is operated by the cam *M* through the lever *L*. *S* is the igniter which was referred to in a previous paragraph.

MULTICYLINDER STATIONARY ENGINES

The multicylinder vertical engine has become very popular for small power plant and other industrial purposes. Because it is multicylinder the torque on the shaft is more uniform than when a single cylinder is used. Also, having a number of cylinders, each cylinder is small for a given power compared to the single-cylinder type and higher rotative speeds may be used. Again, vertical engines require less floor space than horizontal engines, and they may be more easily balanced to avoid disagreeable and destructive vibration.

The four-cylinder engine manufactured by the Novo Engine Company for use with industrial equipment such as hoists, pumps and air compressors for construction purposes, is shown in Fig. 79. The complete unit including radiator is mounted on a cast-iron base, the bottom of which forms the fuel tank. Lubrication is effected by a combined splash and force feed system. The valves are located in the head which is detachable. The crank-

shaft, which has three bearings, carries two flywheels, one at either end.

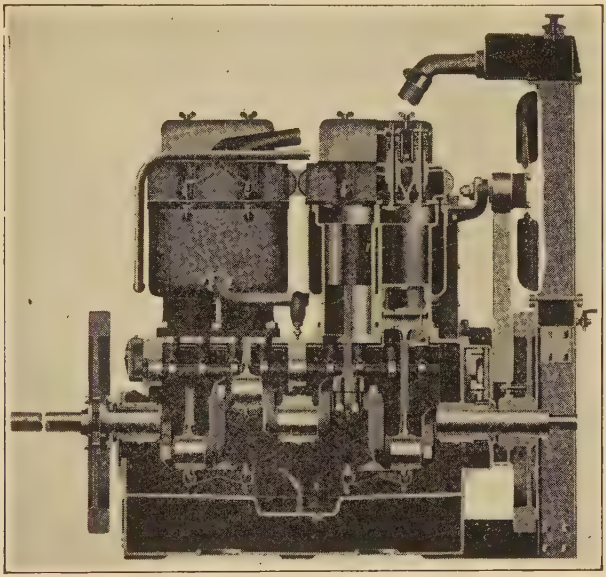


FIG. 79.—"Novo" Four-cylinder Industrial Engine.

This engine is built in three sizes as shown in the table below.

Bore and stroke, inches	Speed, r p.m.	Weight, pounds	Height including radiator, inches	Length, inches	Flywheel diameter, inches
$3\frac{1}{4} \times 5$	850–1000	700	46	36	17
$4 \times 5\frac{1}{2}$	750– 900	1100	50	43	19
$4\frac{3}{4} \times 6$	650– 800	1500	56	57	22

The Sturtevant gasoline engine for small lighting plants is shown in Fig. 80. The design shows the influence of the automobile engine and the characteristics which the two have in common help to make the stationary unit more popular. The automobile engine is reliable and is understood by a large number of people who know nothing about any other type of gas engine. For that reason, a stationary engine, which is to be used largely

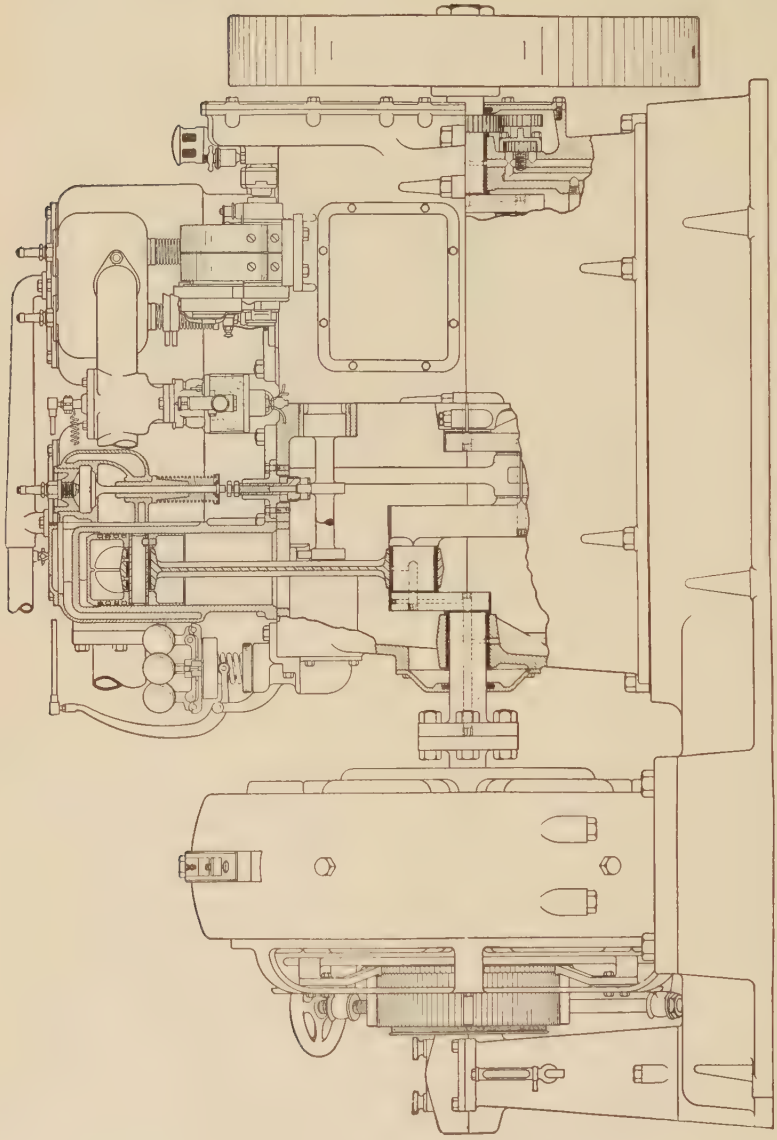


FIG. 80.—Four-cylinder Gasoline Engine. Direct-connected to Direct-current Generator. B. F. Sturtevant Company.

by men who own and are familiar with automobiles, should conform to the accepted general outlines of the automobile engine.

The Sturtevant Company manufactures this engine in three sizes, 5, 10 and 15 kilowatts capacity. The smallest size has 4 cylinders cast en bloc, $3\frac{1}{2}$ inches bore, 5 inches stroke. The 10-kilowatt engine has 4 and the 15-kilowatt 6 cylinders, 4 by 6 inches, cast in pairs. The speed of the smallest machine is 900 and of the two larger, 750 revolutions per minute.

The **cylinders** are of cast iron, the 5-kilowatt unit having the valves all on one side while the two larger units have the inlet- and exhaust-valves on opposite sides. In all three sizes the water jackets are cast integral with the cylinders.

The **pistons** are of the same quality of cast-iron as is used in the cylinders, and are provided with four rings.

The connecting-rods are of I-section and drop forged. The crank-pin ends are of the marine type and are lined with interchangeable die-cast bushings of Parsons' white brass. The piston-pin ends are bushed with phosphor bronze.

The **engine bases** of the 10- and 15-kilowatt sets are made of two iron castings, split horizontally on the center line of the crank-shaft bearings. A single casting is used for the base of the 5-kilowatt set, and in this case the generator is attached directly to the engine base. Separate sub-bases holding engine and generator are provided in the two larger sizes. The crank-shaft is carried on two, three and four bearings in the 5-, 10- and 15-kilowatt sizes, respectively. These are lined with Parsons' white brass interchangeable die-cast bearings.

Lubrication is accomplished by means of a forced-feed system. A gear pump, located in the base of the engine, furnishes oil under 20 pounds pressure to all bearings. The oil enters the main bearings and flows through the crank-shaft, which is drilled for this purpose, to the crank-pins, whence it passes up the connecting-rods to the piston-pins. The oil thrown off from the crank-shaft by centrifugal action serves to lubricate the piston in the cylinder. Oil under pressure is also supplied to the cam-shaft and the governor bearings. The oil falls back into the base, where it passes through a filter and is used over again.

The **governor** is of the centrifugal type and operates by throttling the mixture. Regulation is close enough so that the cur-

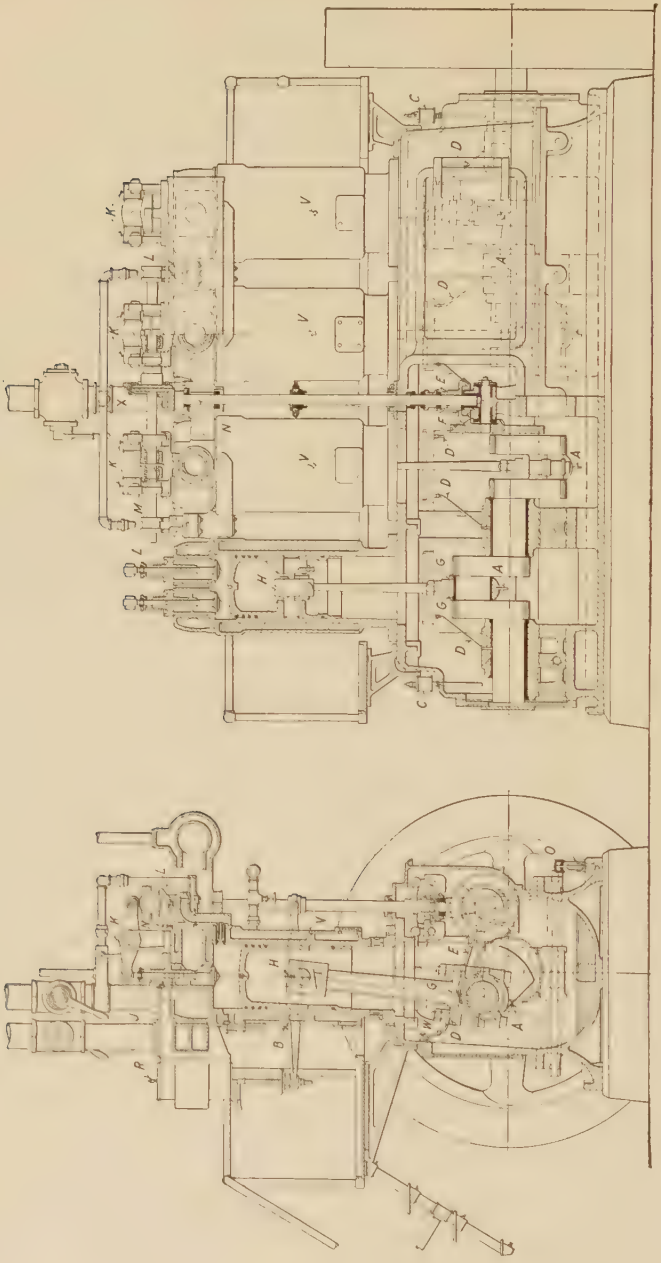


Fig. 81.—Transverse Section.

Fig. 82.—Longitudinal Section and Elevation.

Bruce-Macbeth Four-cylinder Vertical Engine.

rent may be used directly in lighting without the use of storage batteries to maintain constant voltage.

Cooling is accomplished by means of a circulating pump driven from one of the cam-shafts. A tank is furnished so that water may be pumped around through a closed system, the tank acting as a reservoir and a cooler.

Ignition is by means of a Bosch high-tension magneto driven from the cam-shaft. No batteries are used, the magneto being sufficient for starting.

The Bruce-Macbeth Engine.—This engine is illustrated by sectional drawings in Figs. 81 and 82. It will be noticed that the cylinder and jacket are cast together, with head cast separately. The **valves** are in the head, both inlet and exhaust being vertical. They are operated from cams on the cam-shaft which is shown on the exhaust manifold. This **cam-shaft** is driven by means of a pair of spur gears and two pairs of bevels. On the vertical shaft inside the crank-end is located the **governor.** The sleeve of the governor extends above the governor proper and is connected at its upper extremity to a rocker-arm which passes between the cylinders and is attached to the governor-valve illustrated in Fig. 56.

The **crank-shaft bearings** are adjusted by a wedge on a long threaded bolt which passes out through the crank-case for convenience in making adjustments. The crank-shaft and crank- and wrist-pins are oiled by a splash system, the piston by a forced feed. The cam-shaft, it will be noticed, is oiled by ring oilers.

The Foos Engine.—The three-cylinder model of the Foos Type V engine is illustrated in Figs. 83 and 84. The same type is built also in two- and four-cylinder models. Units range from 35 to 325 horse-power. The valves of this engine are located in the head and each valve is driven from a separate cam on the cam-shaft, which is located in the base. Push rods from cam rollers operate rocker-arms on the cylinder heads, which in turn open the valves. The cams are made in half to allow removal of any cam without disturbing the cam-shaft.

The pistons are equipped with five rings at the top and two at the bottom. The rings are eccentric and cut with square lap joints.

The connecting-rods are of open-hearth forged steel and are rectangular in section. The design of the rod is such that the piston-pin bearings can be adjusted by keys placed in line with the large openings in the rear of the crank-case.

The main-bearings are cast separate from the base and turned to fit the corresponding bore of the base. Each bearing is removable without disturbing the shaft. Adjustment for wear is provided by liners under the cap.

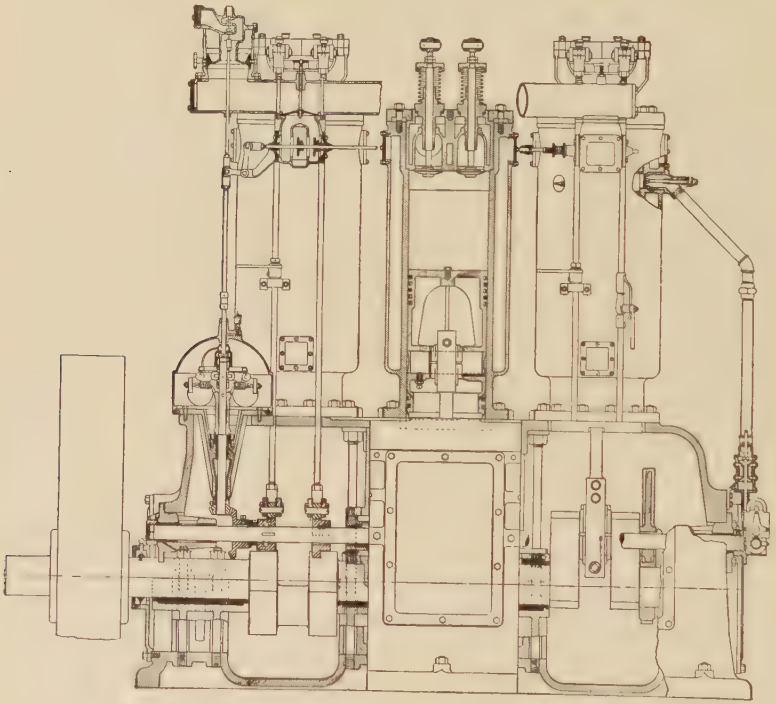


FIG. 83.—Foos Three-cylinder Vertical Engine.

The governor is of the gear-driven centrifugal type and operates on the mixer valves and also controls the double poppet, balanced throttling valves located at each cylinder head. The governor gears are inside the base.

The ignition on the three- and four-cylinder units consists of two independent high-tension jump-spark sets geared direct to the engine as shown in Fig. 84. These magnetos are of the impulse

type so that no current from an outside source is required in starting the engine. A single high-tension set is furnished on the two-cylinder unit.

The lubrication of the main bearings, connecting-rod bearings, cylinders, pistons, cams and gears is accomplished by the splash

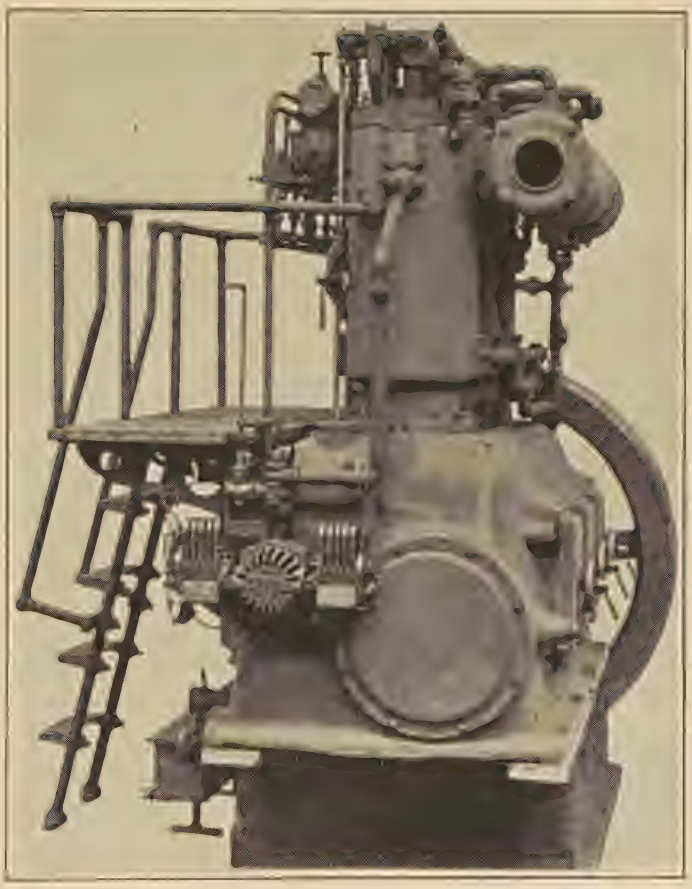


Fig. 84.—End View of Foos Three-cylinder Vertical Engine.

system. Pockets and funnels assist in catching oil and conveying it to the crank- and wrist-pins.

The engine is started by compressed air working on every downward stroke of one cylinder until the other cylinders take up their regular operation.

The latest type of Foos engine, known as Type T, is shown in Figs. 85 and 86. It is built in units of two to six cylinders. One of the distinctive features is that all moving parts including the flywheel and governor are enclosed but nevertheless very accessible. The cam-shaft may be removed from the front side completely assembled. Every element of the cam shaft assembly, *i.e.*, bearing caps, cams, levers, etc., are removable without disturbing any other part. The crank-shaft is removable

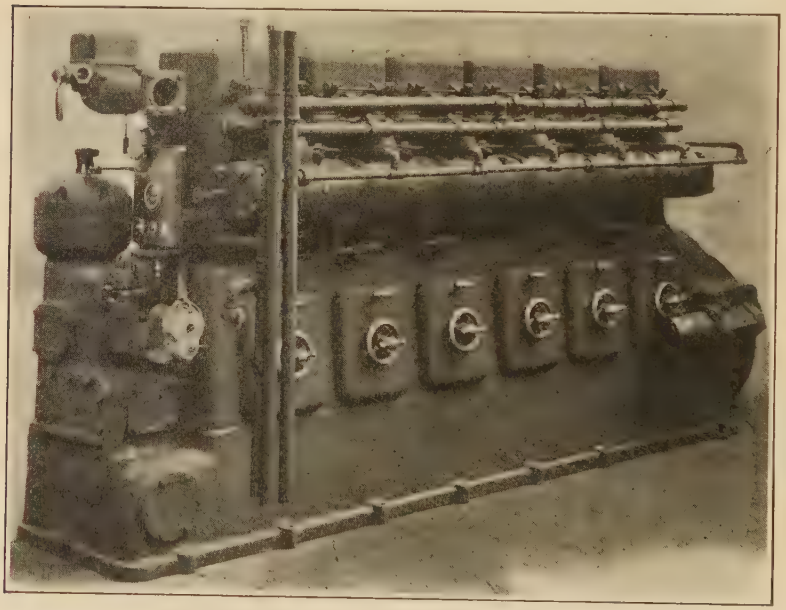


FIG. 85.—Foos Six-cylinder Type T Engine.

from either side or end of the engine. Cylinder heads may be removed without interfering with either manifold. Pistons and rods may be taken out by removing the head only. The governor may be taken out as a complete unit as well as the oil pump.

The valves are located in the heads.

The pistons are of the usual trunk type and are made as light as possible. The length of the pistons is about equal to twice the bore. The piston pin is of alloy steel and made hollow.

The connecting-rod length is $2\frac{1}{4}$ times the stroke. The piston-pin bearing is a bronze bushing and the crank-pin end is of the marine type with four chrome nickel steel bolts.

The crank-shaft is made of .35 to .40 carbon open-hearth steel. The flywheel and power drive are bolted to flanges forged integral with the shaft. There is a bearing on either side of each crank.

Fig. 86.—End View of Foos Type T Engine.

The total projected area of the main bearings is 75.25 square inches in the two-cylinder unit and 190 square inches in the six-cylinder.

Ignition is by means of a high-tension magneto.

CHAPTER XIII

AUTOMOBILE ENGINES

The automobile engine is distinguished from all other internal combustion engines and indeed from all other prime movers by its capacity to develop anything from 5 to 90 per cent of its maximum indicated mean effective pressure or torque over a speed range of 100 to 3500 r.p.m.

It is also capable of a sudden transition from high speed and low torque to low speed and maximum torque by the mere pushing of an accelerator pedal sometimes supplemented by the operation of the spark timing device.

It is the striving to this end which has brought into being the large number of types of engines in common use. Thus we find four-, six-, eight- and twelve-cylinder engines used without any particular classification as to price or the character of the automobile in which they are used.

Within recent years the technique of engine design has progressed beyond the cut and try stage although it is possibly more influenced by public non-technical opinion than by any other single consideration. The number of cylinders an engine possesses is supposed to connote in some mysterious fashion the power it can produce and to be directly related to its hill climbing capacity, flexibility, etc. There is, of course, a germ of truth in this belief as in most popular beliefs but only when interpreted very strictly. Thus the power developed by an engine is a function of the piston displacement, mean effective pressure, and revolutions per minute. If the revolutions per minute and mean effective pressure are postulated there is nothing to choose between a few cylinders of large individual capacity or many cylinders of the same total capacity so far as brake horse-power is concerned. The automobile engine, however, is not judged by its power performance alone. It must deliver its power silently and smoothly to the chassis in which it is mounted so that the passenger may not suffer discomfort. For this reason an engine

must be in good reciprocating balance and deliver its torque without appreciable lack of continuity. These effects for a given engine speed and torque are dependent upon the size of the individual cylinders and the number and arrangement thereof.

In automobile engine design there are few problems less reduced to scientific precision than that of the factors which implicitly or explicitly determine the number of cylinders which shall be used for any particular car. In a way, of course, the problem is indeterminate being controlled by subjective and indeed physiological considerations but nevertheless the general principle at issue should be grasped.

So far as reciprocating balance is concerned all engines may be placed on a common footing. Six- and twelve-cylinder engines are inherently in perfect balance; two-, four- and eight- and even single cylinder engines may by the use of simple mechanism such as the Lanchester anti-vibrator be put in perfect reciprocating balance.

Unfortunately this does not appreciably affect the matter of continuity of torque or in the language of the layman the "smoothness" of an engine.

Action and reaction being equal and opposite it follows that the torque delivered by the crank-shaft to the flywheel must be reflected back by the connecting-rods and pistons to the cylinder walls and thence to the engine supports and the chassis frame.

The use of a flywheel enables the propulsive torque to be delivered to the clutch and chassis mechanism with any desired degree of continuity and there is no practical difficulty in making the torque variation so small as to be unnoticed by the driver or passenger.

The torque reaction, *i.e.*, the varying impulses upon the engine supports are not dealt with so easily.

The number of impulses so given to the engine supports per unit of time are proportional to the number of cylinders assuming that these are all working on the four-stroke cycle.

The magnitude of the said impulses depends upon:

1. The torque developed per cylinder.
2. The number of cylinders as affecting the difference between maximum to minimum torque.

The "apparent intermittence," *i.e.*, the degree to which the senses perceive the torque reaction varies to a first approxima-

tion directly as the difference between the maximum and minimum torque and indirectly as the number of impulses per unit time. Now torque is in itself but a means to an end namely that of propelling a car. The performance in terms of acceleration, hill climbing and speed depends upon the class of car under discussion, but in order to sell a car at all it must have the capacity to accelerate and climb some sort of gradient without recourse to gear shifting. In this respect cars are extraordinarily uniform regardless of make at speeds under 25 m.p.h. a very common acceleration being some 2.5 to 3 feet per sec. per sec.

If such figures are taken as being generally acceptable it will readily be seen that the tractive effort required simply depends upon the total weight of the car and becomes equal to that required to overcome rolling friction say 40 pounds per ton plus the total weight of the car multiplied by $\frac{2.5}{32.2} = 180$ pounds per ton or some 220 pounds per ton in all.

This value of tractive effort is that required at full throttle or maximum torque to produce the desired acceleration of 2.5 feet. per sec. per sec.

The gear ratios in general use on automobiles vary from 4 : 1 to 5 : 1. The number of impulses due to torque reaction from engines with a given number of cylinders at any given car speed, therefore, only vary by 20 per cent over a wide range of cars. On the other hand the total weights of cars in general use varies by hundreds per cent. If all cars had the same rear axle gear ratio the frequency of the torque reactive impulses at any given car speed would be clearly proportional to the number of cylinders while for the same performance in terms of acceleration and hill climbing the cylinder capacity should be in proportion to the car weight if the torque is considered, as is approximately true, as being proportional to cylinder capacity.

In so far as the number of cylinders is multiplied so is the torque produced by an individual cylinder reduced. Further in so far as the crank-shaft arrangement is to spread these impulses evenly over the two revolutions of the four-stroke cycle so is the ratio of maximum to minimum torque and the difference between them diminished. Typical full throttle torque or crank effort diagrams are shown in Fig. 87, for four-, six-, eight- and twelve-cylinder engines of the same total cylinder capacity respectively.

The difference between maximum and minimum torque is indicated on the diagram for the engines mentioned.

These figures are for engines of 300 cubic inches total capacity having the same compression ratio, and connecting-rod crank ratio. For any other capacity the difference between maximum and minimum torque is simply proportional to the cylinder capacity for engines of the types mentioned and of similar design.

The effect of multiplying the number of cylinders is, therefore, not only to increase the number of torque reactive impulses per unit of time which in itself renders them less noticeable to the occupants of a car but also to reduce their magnitude.

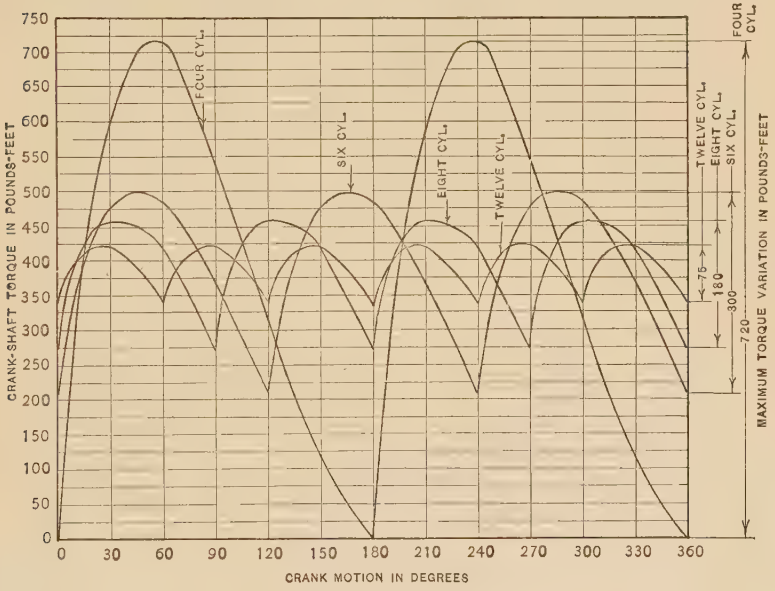


Fig. 87.—Differences in Torque.

The inertia of reciprocating parts is neglected in preparing this figure since it is under conditions of full throttle and low speed that torque intermittence is most serious.

In applying this reasoning to practical design it is necessary to bear in mind that a point is arrived at when further reduction of apparent intermittence is not called for.

Just as a frequency of under 25 cycles per second in an alternate current incandescent lamp produces a most annoying flicker

which is completely remedied by increasing the frequency to 45 or 60 so is it unnecessary to attempt a reduction of apparent intermittence below the point at which it is not physiologically noticeable. Experience has shown that a car weighing some 5000 pounds with a six-cylinder engine of 400 cubic inches capacity developing a brake m.e.p. of 90 pounds per square inch and a gear ratio of 3.6 : 1 will satisfy the most exacting of purchasers in these respects.

From this it can be said that a passable apparent intermittence is proportional to

$$\frac{\text{Difference between maximum and minimum torque per cylinder at full throttle}}{\text{number of cylinders} \times \text{gear ratio}} = \frac{400}{6 \times 3.6} = 18.5$$

A six-cylinder car with an engine of 240 cubic inches capacity gear ratio 4: 1 and the same ratio of car weight to engine capacity as the above, *i.e.*, weighing 3000 pounds would have a torque intermittence of

$$\frac{240}{6 \times 4} = 10$$

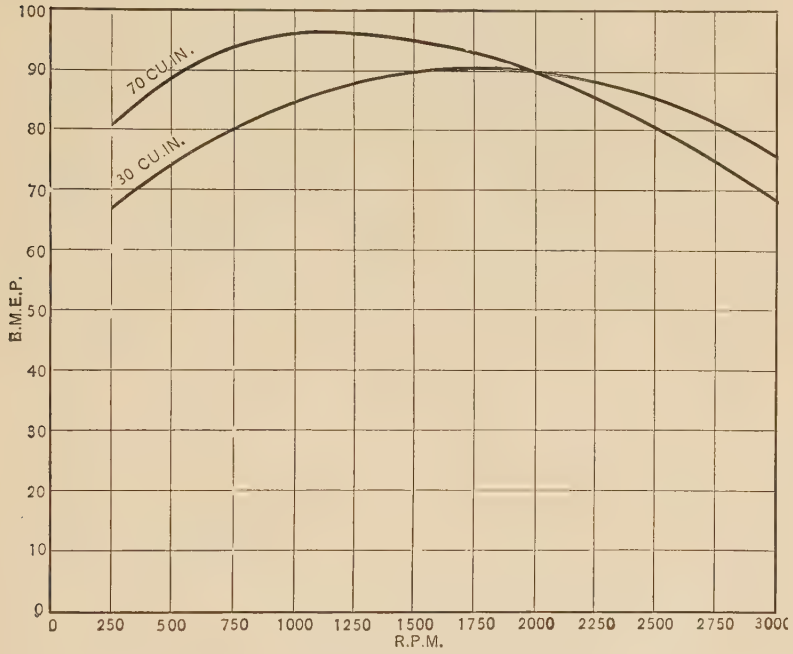
showing how a lower gear ratio and smaller engine profoundly affects smoothness of running.

In deciding the number of cylinders with which to endow a new automobile a calculation along the lines of the above will at least bring the subject within the realm of reason rather than of blind choice.

As gear ratios, *i.e.*, engine speeds, do not vary by more than about 25 per cent irrespective of the make and type of 90 per cent of cars and as the requirements in respect of acceleration and hill climbing vary but little except in the extent to which they can be maintained at high road speeds, in which case even a four-cylinder engine makes an excellent showing, the principal determinant of the number of cylinders is the total weight of the car so that while heavy cars are logically provided with six- and eight-cylinder engines to secure an acceptable smoothness of operation the same smoothness can be obtained in a light car with a four-cylinder engine.

From the diagrams given it will be seen, however, that the effect of increasing the number of cylinders is to reduce the apparent intermittence out of all proportion to such increase.

The choice of the number of cylinders, however, is further determined by the fact that if the capacity of the individual cylinder is very small the cooling losses at low engine speeds are excessive, *i.e.*, the engine will not "pull" at low speeds, whilst



VARIATION OF BRAKE MEAN EFFICIENCY PRESSURE WITH ENGINE SPEED FOR CYLINDERS OF 30 & 70 CU. IN. CAPACITY

FIG. 88.

if the individual cylinder capacity is too large high engine speeds, *i.e.*, a high ratio of maximum to minimum speed, cannot be realized owing to inertia stresses and bearing pressures in the moving parts such as the piston, connecting-rod and valve gear.

The designer is, therefore, forced to take into account many other considerations than would appear at first sight. As a general rule individual cylinders are not less than 30-cubic inch capacity nor more than 70. The former size enables an engine

to be built capable of maintaining its torque effectively up to 3500 r.p.m. but with a torque below average at say 500 r.p.m. while the latter size will have an upper limit of speed at about 2500 r.p.m. but have good pulling capacity at speeds down to 200 r.p.m.

Figure 88 shows typical brake mean effective pressure revolutions per minute curves for cylinders of 30 and 70 cubic inch respectively.

These are based upon the assumption that the reciprocating parts are of conventional design and material. It will be shown later that the use of light alloys for these parts may easily eliminate the speed limitation now imposed upon the large cylindered engine when ordinary materials are used.

Whether four, six, eight, or twelve cylinders are used and independent of their size there are general considerations explained later which enable the designer to predict with fair accuracy the power and speed curve of his engine before it is built.

Packard Single Six Motor.—Figure 89 shows a sectional elevation of a Packard Single Six motor. The cylinders are cast in a block of six and bolted to an aluminum crank-case. The exhaust and inlet valves are of the poppet type and placed on one side of the motor. The water circulating pump and fan are combined and a thermostat is fitted which only allows the water to circulate when the water jackets are at a predetermined temperature of about 160° F. This provision obviates considerable waste of gasoline which occurs when a motor is run at a low temperature. The oiling system is of the full force type, oil being led from the gear pump shown at the bottom of the oil pan to the crank-shaft bearings and from thence through a drilled crank to the connecting-rod bearings.

Lubrication of the wrist-pins and cylinder walls is by a splash from the oil out of the crank-shaft and crank-pin bearings. The valve-operating mechanism is by means of rocker levers and plungers, the function of the rocker lever being to avoid any side thrust and consequent noise between the tappets and their guides. A point of great interest in this motor is the use of a fuelizer between the carburetor and the inlet valves. This device is virtually a bypass from the inlet manifold through which gasoline vapor passes. A sparking plug connected with the ignition ignites this gas so that a very small quantity of gas at a very

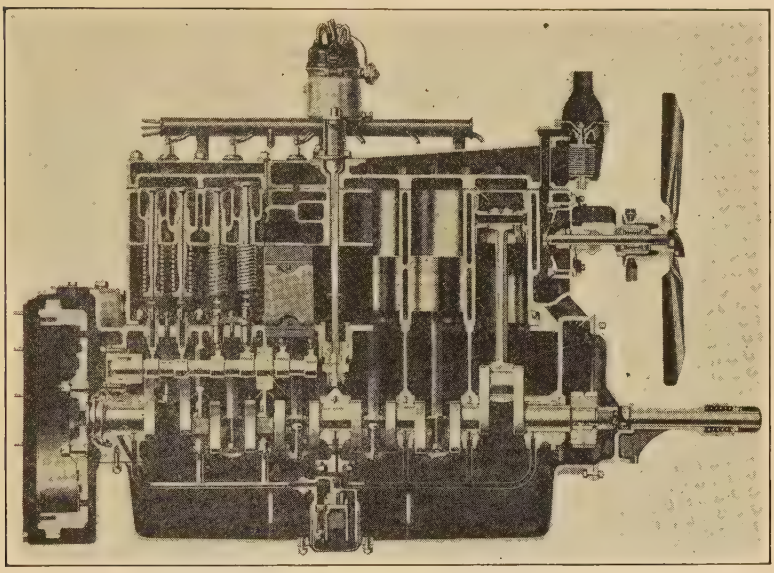


FIG. 89.—Packard Single Six Motor.

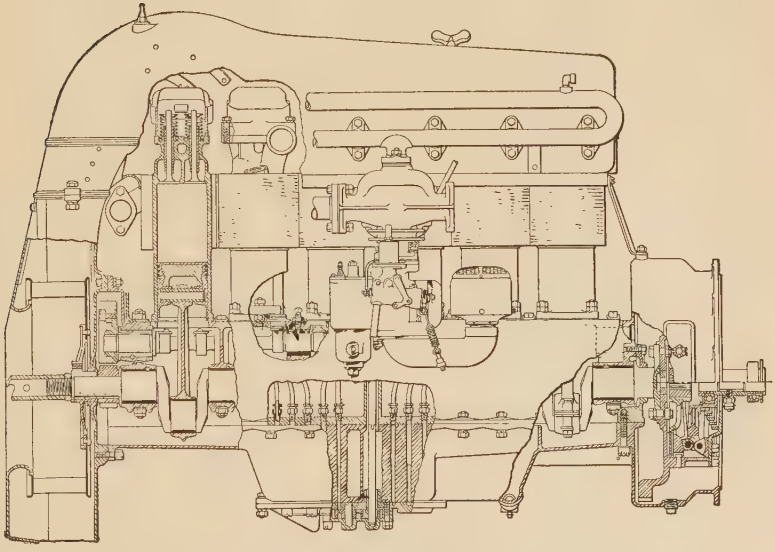


FIG. 90.—Franklin Six-cylinder Automobile Engine.

high temperature is fed into the gasoline air mixture passing through the inlet pipe to the cylinders. In this way very complete vaporization is ensured and good economy obtained.

Franklin Motor.—The motor build by the H. H. Franklin Company, Syracuse, and used in their cars is shown in Figs.

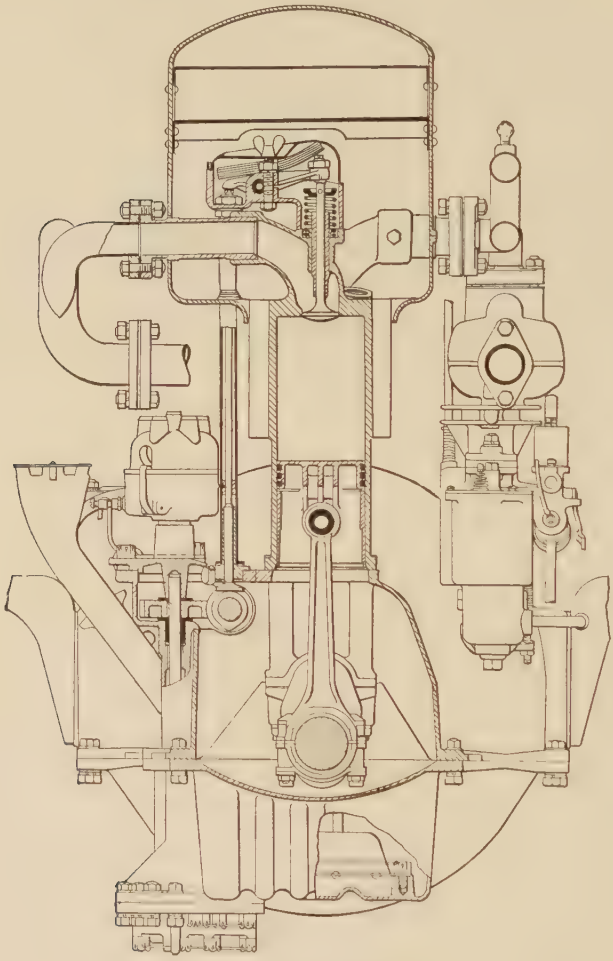
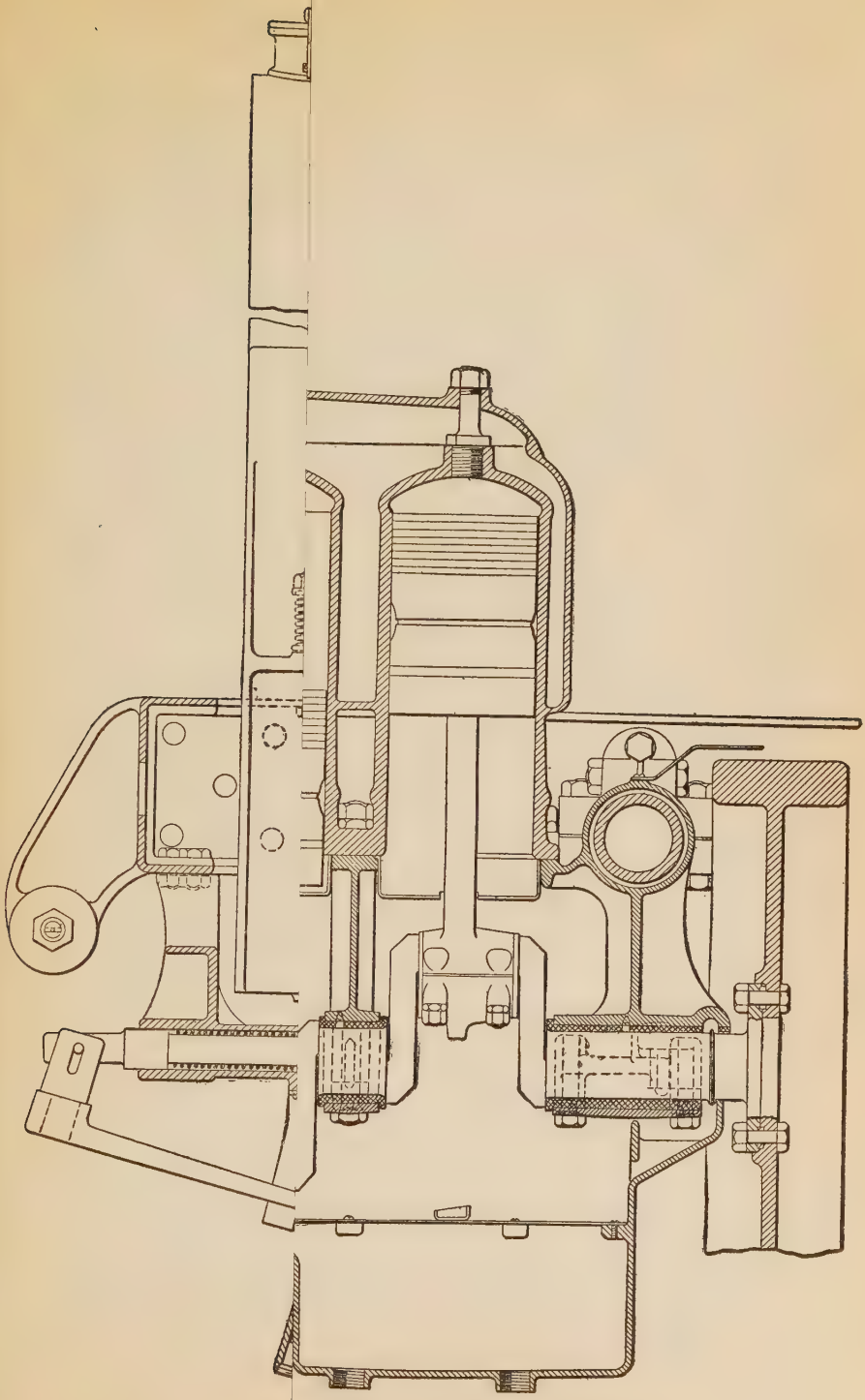


FIG. 91.—Transverse Section Franklin Automobile Engine.

90 and 91. The distinctive feature of this motor is that it is air cooled. The cylinders are cast individually as shown with steel fins parallel to the cylinders cast into place. Surrounding these



(*Facing page* 168)

fins is an aluminum casing which at the forward end of the engine carries a blower. This blower is driven from the crank-shaft and delivers air to the cylinder cooling fins. The valves are of the overhead type operated by vertical push rods and levers. The oiling system is of the forced feed type, the oil being delivered by a vertical rotary pump from the bottom of the crank-case as shown. It will be noted that this pump is of the distributor type, *i.e.*, the oil pressure is applied to each bearing in turn and not simultaneously as on the Packard motor previously described. The object of this arrangement is that the breakage of one pipe or failure of one bearing does not involve failure of lubrication to the remainder of the bearings.

Locomobile Truck Engine.—The Locomobile four-cylinder truck engine is illustrated in Figs. 92 and 93. The diameter of the cylinders is 5 inches, stroke 6 inches, and the rating of the engine is 45 horse-power at 900 revolutions per minute.

The cylinders are cast in pairs with a jacket cast integral with them. The thickness of the cylinder tapers from $\frac{5}{16}$ inch at the bottom of the water jacket to $\frac{7}{16}$ inch at the bottom flange. The admission and exhaust ports are 2 inches in diameter, giving gas velocities of 5700 feet per minute with an engine speed of 900 revolutions per minute. Compression is 66 pounds gauge.

The **crank-case** is of government bronze. It is supported from the frame at front and rear by means of steel tubes, which rest and are held down in brackets, bolted to the frames by means of caps, bolts and nuts.

The **crank-shaft,** which is of alloy steel, heat treated, is supported from crank-case by means of caps, studs and nuts. The aggregate length of the five crank-shaft bearings is $18\frac{3}{8}$ inches.

The crank-shaft **gears** are steel drop forgings with spiral teeth, case hardened.

The **cam-shafts,** of steel, hardened and ground, also have five bearings. The cam-shaft **gears** are of cast-iron with spiral teeth.

The **valves** are steel drop forgings, $2\frac{1}{2}$ inches in diameter, with a conical seat of 45 degrees. The lifter guides are of phosphor bronze. The guides for the valves are separate iron castings forced into place and may be replaced when worn.

The **water pump** is of the centrifugal type and makes 1700 revolutions per minute with the motor running 900 revolutions.

The **oil pump** is driven by spiral gears from the exhaust cam-shaft. This pump is mounted and driven inside the crank-case. Oil is fed by pump into main supply pipe from which one pipe supplies oil to the oil-troughs and another pipe supplies it to crank-shaft bearings. Connecting-rods have scoops which dip into oil-troughs. Crank-pins are therefore oiled directly, and the wrist-pin, piston and cam-shaft bearings by splash.

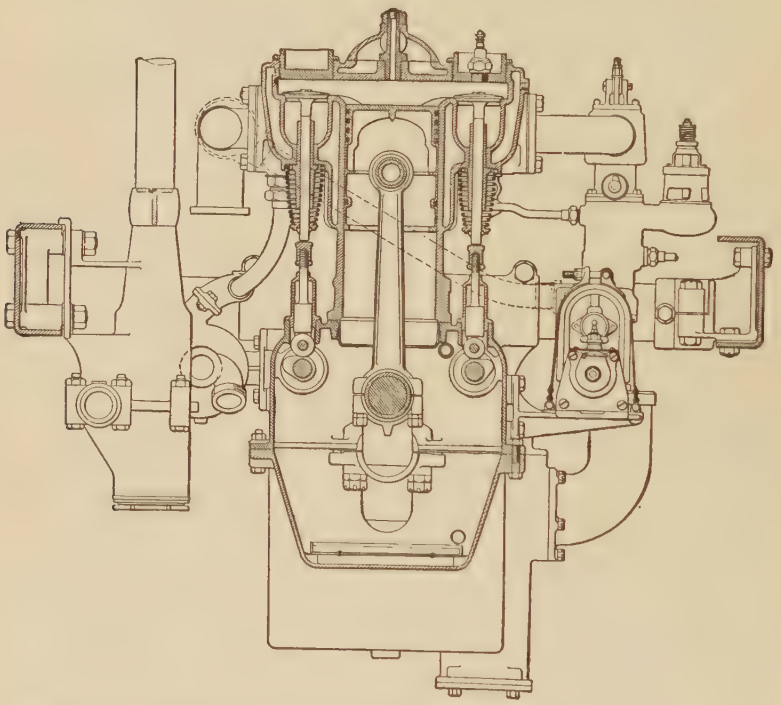


FIG. 93.—Locomobile Four-cylinder Truck Motor—Transverse Section.

Ignition is of the high-tension dual system, magneto and storage battery. The magneto is a Bosch, with coil for starting. The magneto has a fixed spark advance and the battery has a fixed spark at about dead center for starting.

The Stearns-Knight Engine.—There have been many attempts to utilize a valve for the gasoline automobile motor, that would not have any of the disadvantages of the lift poppet valve. The earliest motors used rotating valves but these proved to

be unsatisfactory and poppet valves took their place. Sliding valves of different types have been proposed and some of them experimented with, but none have really proven successful until the Knight sleeve-valve motor was brought out. This motor is illustrated in Figs. 94, 95 and 96. Instead of the poppet valves, the valve action consists of two concentric sleeves sliding up and down between the piston and cylinder walls. These **sleeves** open and close wide slots or ports opening directly into the combustion chamber through which the inlet and exhaust

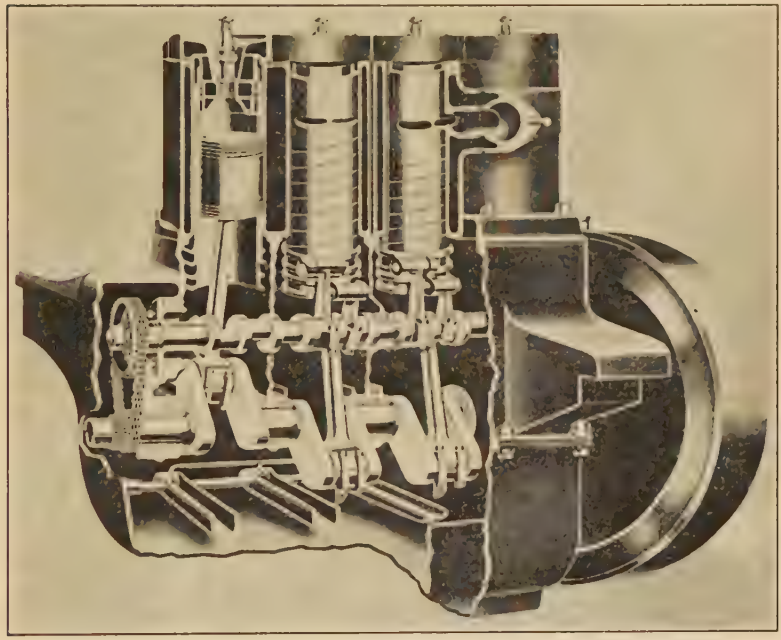


Fig. 94.—Stearns-Knight Four-cylinder Engine—Longitudinal Section.

gases pass. They are moved up and down by small connecting-rods from a crank- or eccentric-shaft in about the same manner as the slide-valves of a steam-engine are driven. This shaft is driven from the main crank-shaft by a silent chain with a 2 to 1 reduction.

The **eccentrics** which drive the two sleeves for one cylinder are set at about 80 degrees. The slots in the sleeves "register" at the proper time to give inlet and exhaust functions, and this

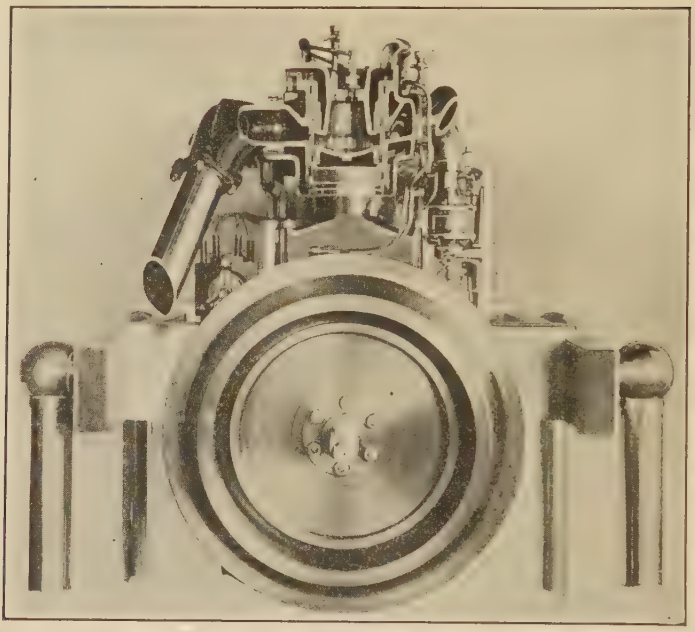


FIG. 95.—Stearns-Knight Four-cylinder Engine—Transverse Section.

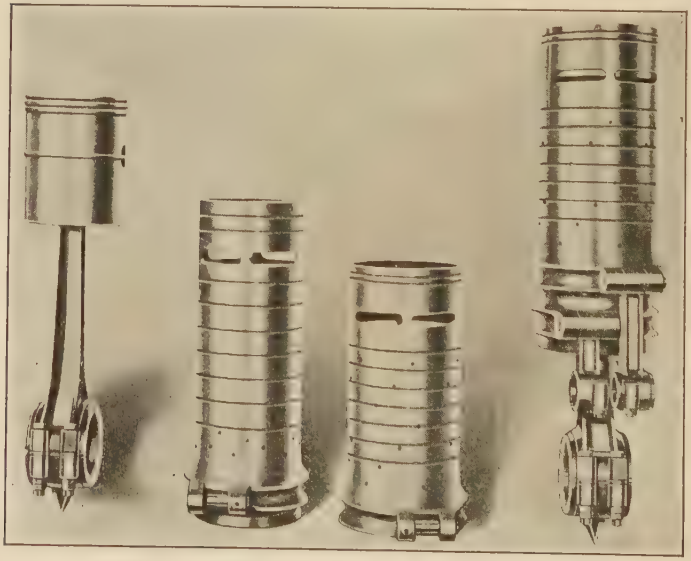


FIG. 96.—Stearns-Knight Engine—Piston and Valve Sleeves.

period of registrations or overlapping of the ports is just when the velocity of the sleeve is a minimum. Thus, the events occur at the proper time, the ports open rapidly, remain open during the required period and then close sharply.

The **carburetor** used on the four-cylinder motor is of the double-jet type; one jet supplies gasoline for slow and intermediate speeds, and an auxiliary jet comes into service automatically at higher speeds.

The **oiling system** consists of an arrangement of movable troughs mounted in the bottom of the crank-case. The troughs are filled with oil at all times, the connecting-rods passing just over them. Scoops placed at the bottom of the rods dip into the oil. The lever actuating the movements of these troughs is connected to the throttle and as the throttle is opened wider and the load on the motor increased, the troughs are raised so that a greater supply of oil comes into use. The troughs are each fed from a separate pipe, which in turn is fed by a force feed pump. The pump is gear driven and with all the piping is contained in the bottom of the crank-case. In addition, an auxiliary oil supply is also connected to the throttle, but in such a manner that it becomes operative only when the motor is worked at excessively high speeds or when the load has become very heavy.

Ignition is by the dual system. It consists of an arrangement of the Mea high-tension magneto and a single vibrator coil, whereby a single set of spark plugs affords a double system of ignition. The magneto is also chain driven.

CHAPTER XIV

LARGE GAS ENGINES

General.—It is rather difficult to draw the line between medium-and large-sized gas engines. Medium-sized engines, however, are built in both vertical and horizontal units, while large engines are not built in vertical types. There is another difference that usually exists: large engines are made double-acting, while small- and medium-sized units are almost invariably single-acting for stationary work. We might, if we have no objection to getting down to fine points, conceive of a single-acting engine made double-acting, and thus transferred from the medium- into the large-sized class. It is certain that if we have two engines of the same bore and stroke, one single- and the other double-acting, the latter will be much larger than the former, not only in horse-power but also in weight and floor space required.

In taking up the subject of large engines for discussion, it will be impossible to go into great detail with even a few different makes of engines. It would be well to describe one or two in detail and take up distinctive features of others and this is the plan the author follows. This scheme will leave the reader with a good general idea of the large gas engines made in this country, for there is a similarity that runs through all different makes, and this similarity is more marked in large engines than in smaller ones. The details that differ are usually the valve gear, governing system and details of construction. The latter, it will be impossible to point out in this chapter but they will be touched upon in the design.

The Cooper Engine.—This engine is built in single-tandem and twin-tandem, double-acting units from 200 horse-power up. All units work on the four-stroke cycle. The one shown in Fig. 97 is a single-tandem direct-connected to a natural gas compressor in the rear. The introduction of the high-pressure system of transporting natural gas has opened up a large field for gas engines and many large units have been sold for this purpose.

The Frame.—The frame or bed-plate of this engine is cast in what is known as the Tangye rolling mill design. It is cast

with one bearing only, so that one side of the frame is high, of box section, while the other is lower. The illustration, Fig. 97, shows the low side. The main-frame extends back to the first cylinder, which is bolted to it. The **crosshead slide** is formed on the bottom of the frame, the top being open.

Cylinders.—The cylinders are made of close-grained, remelted charcoal iron. They are symmetrical one-piece castings, but the entire jacket is not cast with the cylinder. There is a ring left out of the jacket in the middle of the cylinder, leaving part of the jacket projecting from each end of the cylinder. This opening

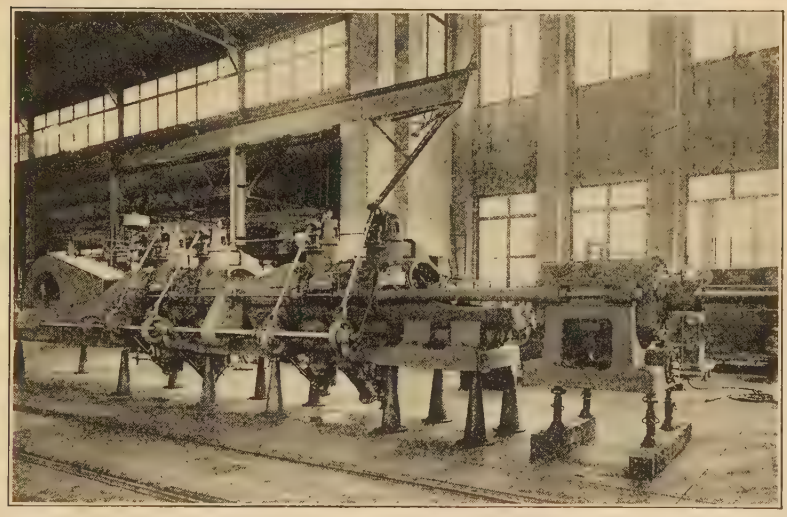


Fig. 97. — Cooper Tandem Double-acting Gas Engine.

is closed by a **jacket band** in three segments. The bolts and flanges of this band may be seen in Fig. 97. The construction is shown in Fig. 196.

The cylinders are supported in the counter-bores of the **main bed-plate, distance-piece** between the cylinders and **tail-guide.** As the **pistons and rods** are carried by the **crossheads,** the cylinders support only their own weight.

Pistons. — The **pistons** are cast in one piece without any internal ribs. Each piston is fitted with three **rings** made in segments, having malleable iron **keepers** at the joints. These packing rings are held out against the cylinder walls by steel

springs. The pistons are forced on the rod against a shoulder and held there by a flush **nut.** Provision is made to insure circulation of water through the piston.

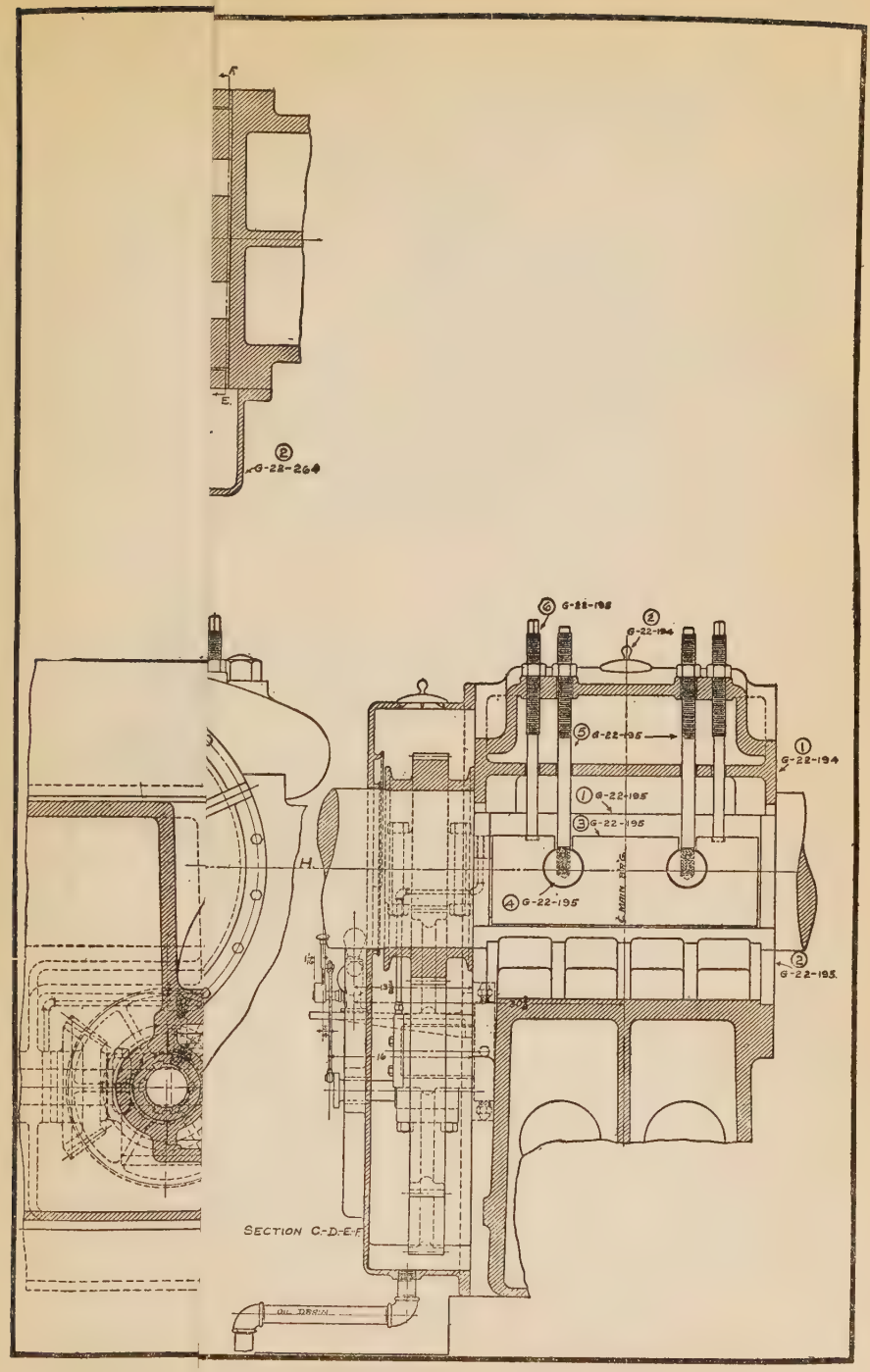
Piston-rods. — The **piston-rods** are nickel steel forgings drilled from both ends for water circulation. They are fastened to the main, intermediate and tail-guide crossheads by nuts. All rods are of the same size and interchangeable. The rods are fitted with **metallic packing** of the sectional type, consisting of segmental cast-iron rings placed along the rod in series. The packing includes two **fire rings** adjacent to each combustion chamber, which are compressed by external snap rings, and the remaining rings in the series are held in contact with the rod by garter springs.

Crossheads. — The **main** and **intermediate crossheads** are of open-hearth steel. The main crosshead is forked, and is connected to a flat babbitted shoe by means of a cylindrical joint, in which vertical and rolling adjustments may be made. The main **crosshead-pin** is of high carbon steel, clamped into both crosshead cheeks.

The **intermediate crosshead,** with its cylindrical shoe, acts as a coupling for the two piston-rods and admits cooling water, which is made to enter at the ends of the rod. The **tail-guide crosshead** also has a cylindrical shoe provided with means for centering and adjustment.

Distance Piece and Tail-guide. — The distance piece between the cylinders rests on machined ways on a **base-plate** to allow for expansion and contraction of cylinders. This casting also provides two bored guides, one for the intermediate crosshead and one for the water crosshead, to admit cooling water to the rods and pistons through tubes that telescope with water chambers. The base-plate underneath supports bearings for a lay-shaft. The **tail-guide** is a cylindrical casting and provides a slide for the rear crosshead. This casting is cored out to allow the escape of cooling water from the rear piston-rod.

Connecting-rod. — The **connecting-rod** is made of open-hearth steel and is of solid end construction as shown in Fig. 175. Both ends are provided with babbitt-lined shells and wedge adjustment. The adjustment at the crank-pin end tends to shorten the length of the rod while the adjustment at the other end tends to increase the length.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1 G-22-195
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H
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SECTION C-D-E-F
OIL DRAIN

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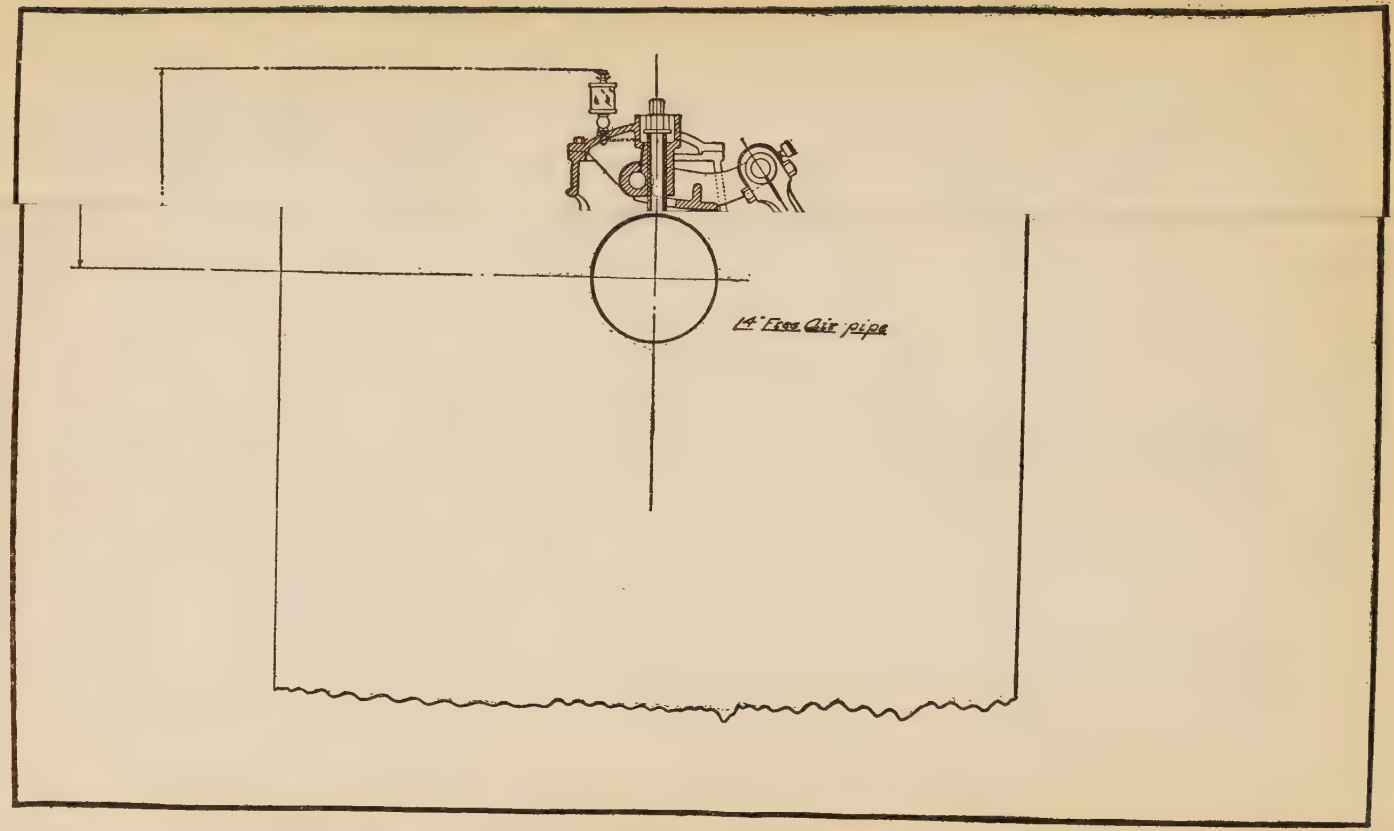


FIG. 99.—Section Through Combustion Chamber of Cooper Gas Engine.

(Following FIG. 98.)

Crank and Main-shaft. — The side-crank construction is used in this engine, as in practically all large American gas engines and steam engines. The **crank** is a high-carbon open-hearth steel casting having the **counter-weight** and **crank-pin** cast solid with it. The crank is forced onto the main-shaft under hydraulic pressure, and is keyed into place.

Valve-gear. — The **cam-shaft** is driven from the crank-shaft through three spur gears and one pair of bevel gears. These gears are enclosed in a case and run in oil as shown by Fig. 98. The cam-shaft proper is supported on bearing brackets cast on the engine base. It will be noticed that the shaft is held between two collars at the bevel gear. It is free to move longitudinally at all the other bearings, however, — a necessary precaution to allow for expansion and contraction. This precaution is even more necessary when the bearing supports are on the cylinders.

The **valves** are operated by **eccentrics** and rolling levers as shown in detail in Fig. 99. The **exhaust-valve** is water cooled. Both valves have a constant lift. **Governing** is accomplished by rotating the sleeve which encloses the inlet-valve spring. This sleeve has ports which register or cover as the load increases or decreases. These sleeves are under the control of a Jahn centrifugal governor.

This system gives constant-quality regulation with variable quantity. There is a means, however, of adjusting the sleeves by hand in order to get the proper mixture for any kind of gas.

Ignition. — The **make-and-break** system is used and two igniters are used in each combustion chamber as illustrated in Fig. 100. The **cam** which operates the igniters is on the **igniter-shaft** which is above and parallel to the cam-shaft as shown in Fig. 97. As the finger drops off the high point of the cam, it is pulled back into place by the **coil spring** at the finger pivot. This motion is transmitted to the two push rods and by them to the moving electrode. The moving electrode is made of cast-iron, the stationary one of soft steel. Both are insulated from the body of the igniter by lava sleeves and mica washers.

Ignition may be varied on all igniters, at once, by the movement of one central lever, or each igniter may be adjusted individually.

Lubrication. — The engine is lubricated by a **pressure** oiling system in which uniform pressure is maintained at all of the engine oil feeds by a pump under the control of a **weighted**

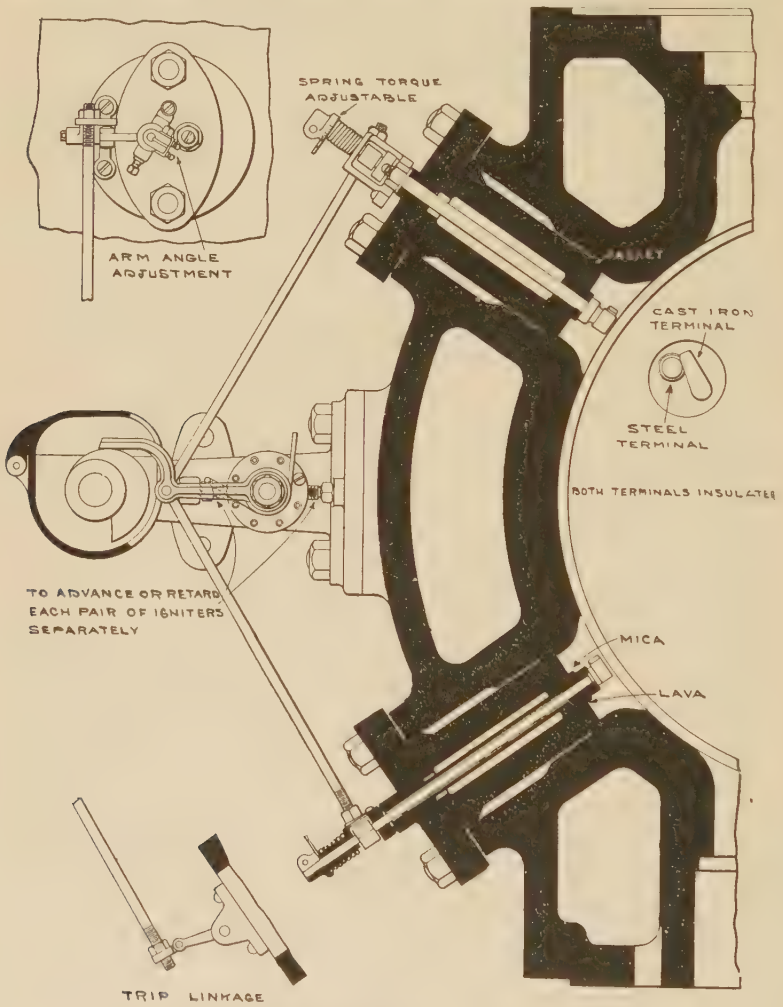


FIG. 100. — Ignition System. Cooper Gas Engine.

valve, which automatically returns any excess oil to the filter through a by-pass. The rotary oil pump, which supplies oil from the filter to the pressure feeds, is driven from the main engine. The oil, after being used, is caught and returned by

gravity to a settling tank at the side of the engine foundation. From this tank the oil flows into a filter, where it is cleaned to make it suitable for using again. The cylinders and piston-rods are lubricated by multiple force feed pumps, driven from the igniter shaft. Oil is admitted to the piston-rods through the metallic packing.

Starting.—The air, gas and water valves are placed at a central point, all within reach of the operator. To start the engine the compressed-air and gas valves are opened. The compressed air "cranks" the engine and after a few revolutions the air is automatically replaced by the explosive mixture when ignition

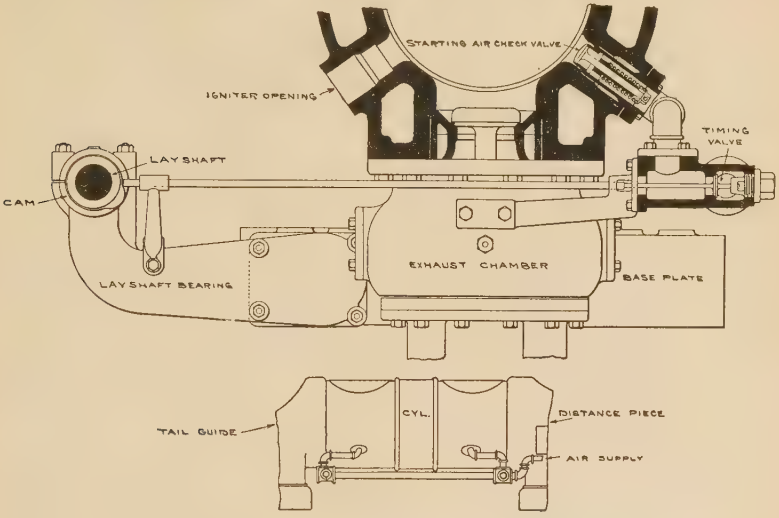


FIG. 101. — Air Starting Gear. Cooper Engine.

begins in the cylinders. This result is accomplished by four cams on the cam- or lay-shaft, each cam operating a poppet valve leading to the combustion chamber of the cylinder. This is shown in detail in Fig. 101. When the compressed-air throttle is opened, air is admitted on top of the **timing valve** and holds it against the seat. The cam forces this valve open at the proper instant, and air is admitted to the cylinder through the check valve. This occurs on the explosion stroke but before explosions begin. When explosions do take place, the pressure in the cylinder is higher than the pressure of the compressed air, the check valve is held on its seat and no air enters. The

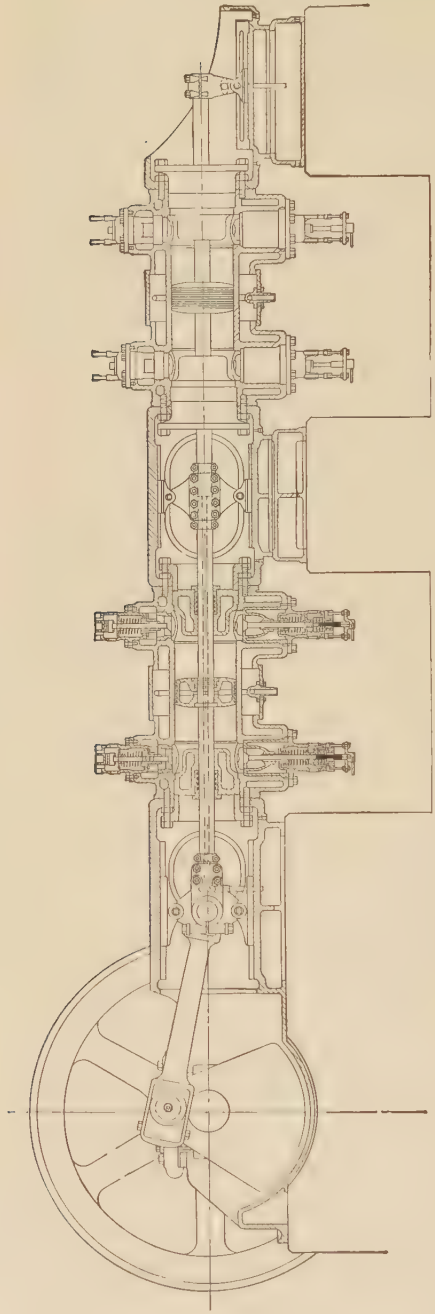


Fig. 102. — Longitudinal Section of Buckeye Four-cycle Engine.

operator then closes the compressed-air valve and the timing valve drops away from its cam.

The Buckeye Engine.—The double-acting tandem four-cycle type of Buckeye engine is illustrated in Fig 102. This type is made in sizes from 250 to 3000 horse-power or in the twin type from 500 to 6000 horse-power. The same company, the Buckeye Engine Co., Salem, Ohio, furnishes single-cylinder double-acting engines, and single, two- and four-cylinder single-acting engines. All are made horizontal. The engine illustrated has cylinders cast with the non-continuous water jacket described before. The different parts of the engine including main-frame, cylinders, distance-piece, tail-guides and supporting bed-plates are clearly shown.

The **pistons** are made in halves, split at right angles to the axis of the rod and clamped together. Six packing rings are used. The **piston-rod** is made in two parts and joined at the intermediate crosshead. Each part is drilled through from end to end. The **main crosshead** is a steel casting, threaded to receive the piston-rod. It is also split and clamped

on the rod by means of through bolts. The crosshead **pin** is straight and is secured in the crosshead by means of clamping bolts. The **shoes** are steel castings and have a swivel connection to the body of the crosshead, and are adjusted for wear by eccentric bolts, which are clamped after adjustment. The **intermediate crosshead** has the same swivel shoe connection as the main crosshead, and also serves as a clamping nut to connect the two sections of the rod. The **tail-rod crosshead** consists of a simple steel casting provided with one swivel shoe and has the same adjustments as the other crossheads.

The **connecting-rods** are forged from open-hearth steel. In double-acting engines the rod is solid at both ends and has bronze boxes at the crosshead end with wedge adjustment. At the crank-pin end the boxes are babbitted and also provided with wedge adjustment. In **single-acting** engines the crosshead end is made solid, but the crank-pin end in this case is of the marine type with babbitted boxes.

Each pair of **valves** on this engine is driven by a single eccentric. The eccentrics are clamped to the **lay-shaft.** They are turned spherical on the periphery and are adjusted by means of a clamped joint. The exhaust valve is water cooled. The lay-shaft is placed in a horizontal plane passing through the center of the engine. It is driven by a set of steel gears, one of which is mounted on a drag shaft that is rotated by the main crank-pin through a universal joint. This drag shaft, which is shown in detail in Fig. 103, also carries a bevel gear which drives the governor. These two pairs of bevel gears are entirely enclosed and are oiled automatically. The drag crank serves for carrying lubricating oil to the main crank.

The **ignition system** is designed to operate with either jump or make-and-break spark. The double-acting engines are provided with two or more igniters in each combustion chamber and can be operated with high- and low-tension ignition simultaneously. The engine is usually equipped with an electromagnetic make-and-break system, made by the makers of the engine. Current from a storage battery, magneto or generator operates the igniter mechanism by means of an iron-clad electromagnetic hammer. The circuit is periodically completed and the time of ignition controlled by a "timer." This timer is provided with three adjustments: First, automatic control di-

rectly by the governor which advances the ignition on light loads when the charge is lean; second, hand adjustment of all igniters simultaneously; third, hand adjustment of each igniter independently while the engine is running.

The **cooling water** enters the piston-rod at the intermediate crosshead and leaves the rod at opposite ends, draining through slots in the bed and tail-rod guide. Each cylinder, cylinder head, exhaust-bonnet and exhaust valve is cooled and has its own independent water circulation.

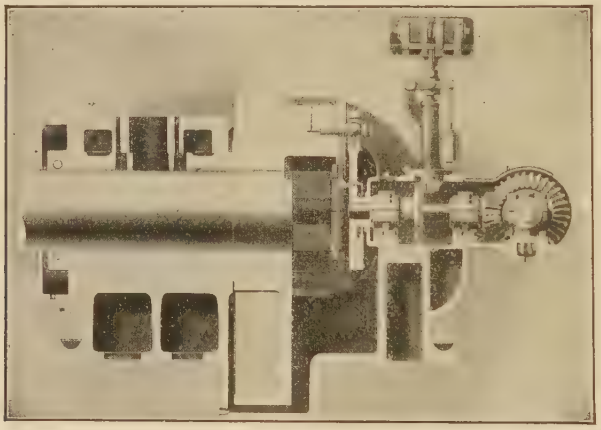


FIG. 103.—Cam-shaft Gears. Buckeye Gas Engine.

The Mesta Gas Engine.—This engine, which is manufactured by the Mesta Machine Co., Pittsburgh, Pa., is illustrated in Figs. 104 to 107. It is of the horizontal four-cycle, double-acting tandem type, arranged either in single or twin units.

The main frame or bed-plate is of the center-crank type having two journals symmetrically arranged about the center line of the engine. The rear end of the frame is provided with heavy double walled hood and double flanges for the purpose of obtaining maximum rigidity between the main frame girders and also with the forward cylinder.

The cylinders are cast in halves and connected in the middle by means of heavy shrink links. The cylinder bore is provided with a liner extending over the entire stroke of the piston.

The crank-shaft is of the center throw built-up type. The crank webs and counterbalances are made of open-hearth steel castings and the crank-pin of a high carbon steel forging. The

crank-shaft sections are of forged steel hollow bored throughout their entire length.

The connecting-rod is of the marine type at the large end and the closed type at the small or crosshead end. It is of forged steel, rectangular in section, with bronze boxes lined with babbitt. Fig. 108 shows a rod for a gas engine with a crank-pin 26 inches in diameter.

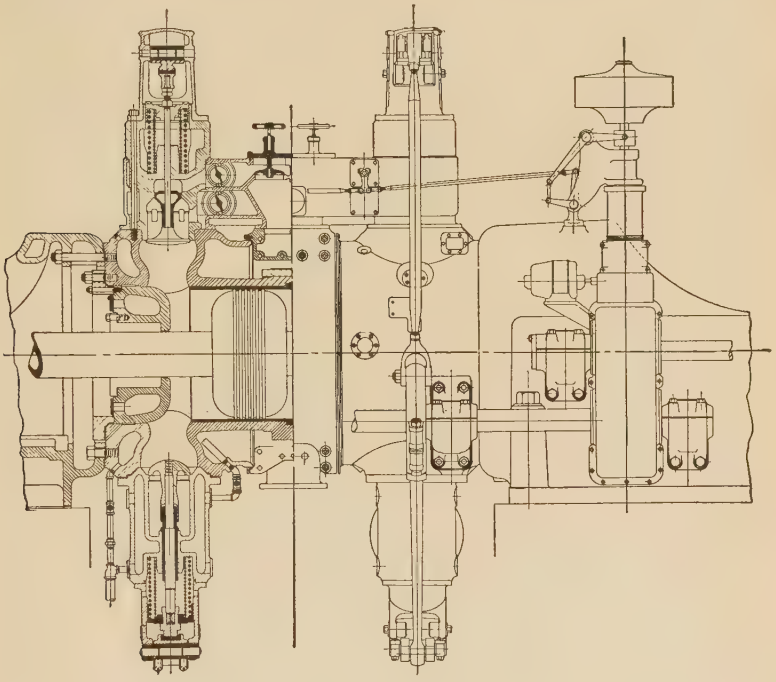


Fig. 104.—Mesta Gas Engine. Section and Elevation.

The pistons are cast hollow and are made in one piece of open hearth cast-steel. They are semi-spherical in shape. The rods are of open hearth forged steel annealed and heat treated. They are hollow bored for water cooling. They are made in two parts and connected at the intermediate cross head. The two parts are interchangeable.

The inlet- and exhaust-valves are both worked from the same eccentric through pull rods and rolling levers. The inlet-valve at the top of the cylinder is of the mushroom type made in one

piece from an open hearth steel forging. The exhaust valve is of cast iron. Both are in removable cages.

The governor is known as the butterfly gear and controls both gas and air by means of butterfly valves placed immediately in front of the main inlet valves. This governor works on the combined quality and quantity regulation in which the ratio of gas to air varies during the higher ranges of load and the ratio of gas to air is kept constant during the lower ranges. The oper-

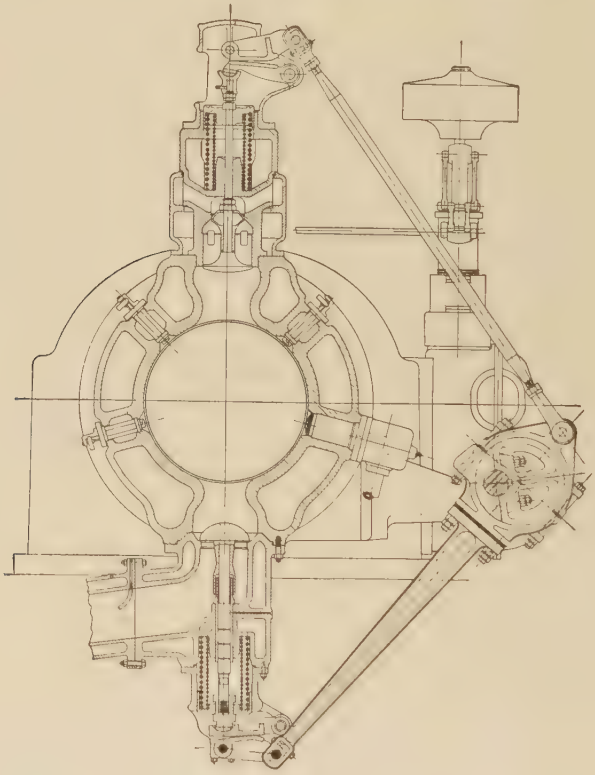


FIG. 105.—Mesta Gas Engine. Section through Combustion Chamber.

ator may, by making a slight adjustment, change the method of regulation to either quantity or quality or any combination of these that occasion may demand.

Ignition is obtained by means of three igniters at each end of each cylinder. These igniters are of the make-and-break type

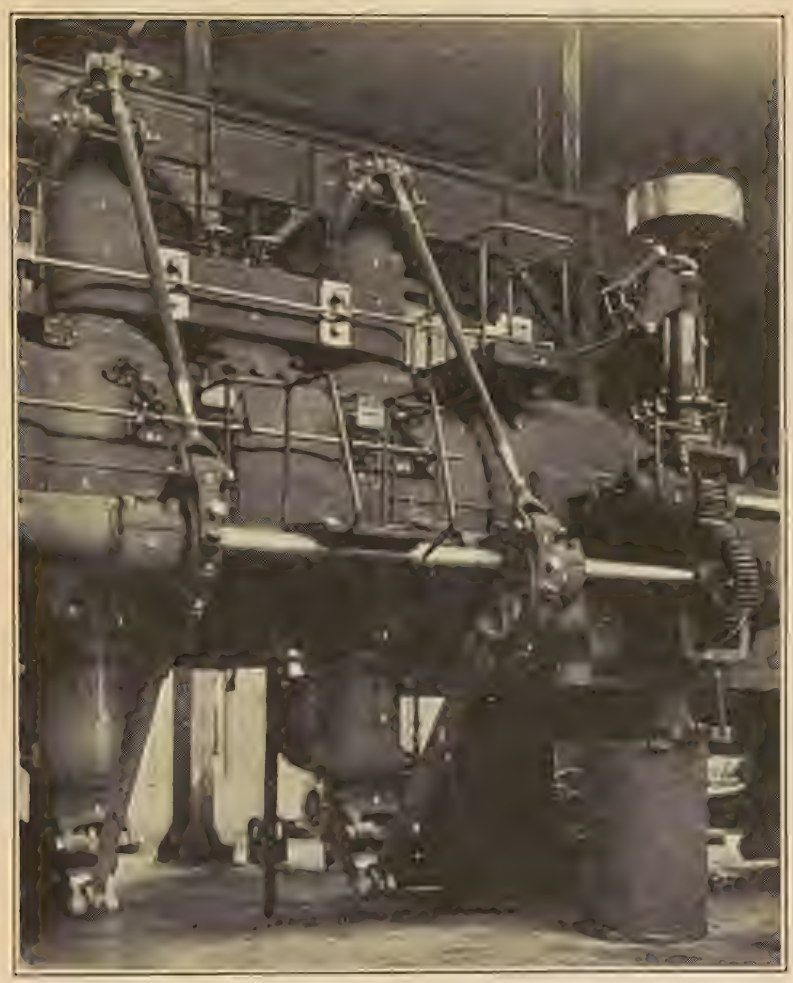


Fig. 106.—Valve Gear and Cylinder, Mesta Engine.

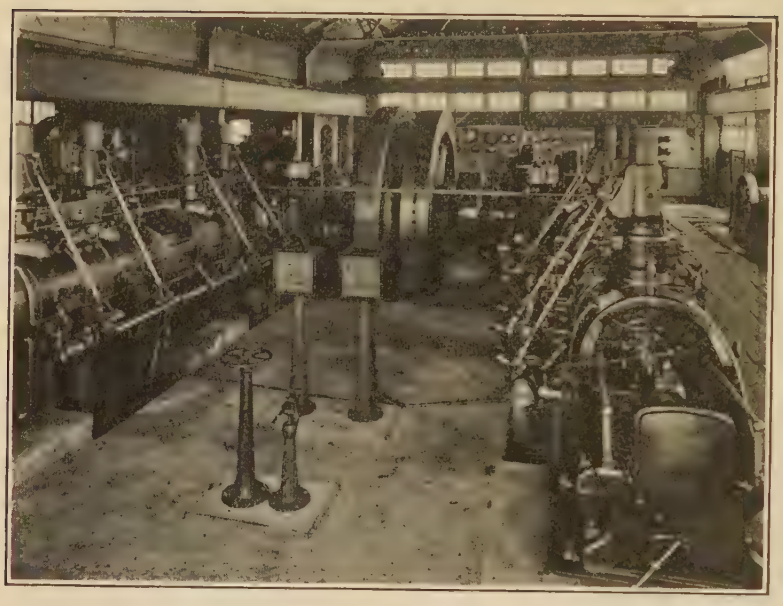


FIG. 107.—Mesta Turn Tandem Gas Engine Direct Connected to Generator.

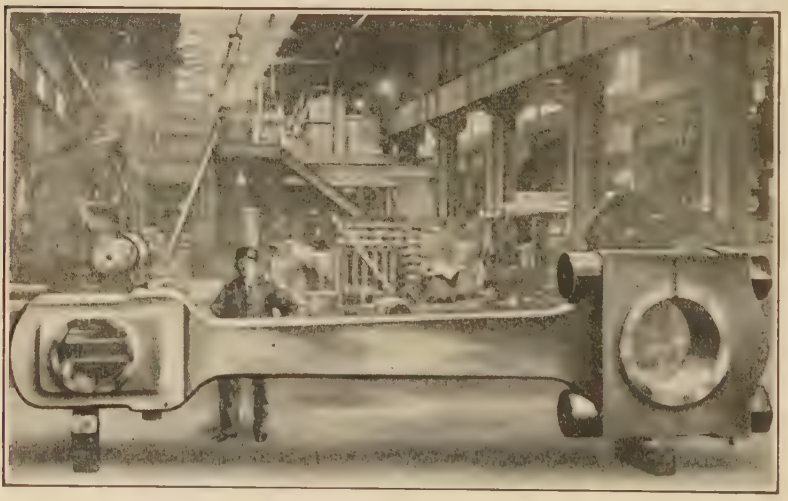


FIG. 108.—Connecting-rod of Large Mesta Gas Engine.

and each has its own circuit with fuses, switch, induction coil and tell-tale lamp.

The air starting mechanism consists of an air starting valve at each end of each cylinder. These valves are operated from the main eccentrics on the lay shaft by a trip arrangement which permits high-pressure air to enter the cylinder at a period which corresponds to the power stroke of the engine. As soon as the engine is up to speed these valves close automatically.

The Allis-Chalmers Engine.—The latest type of double-acting engine built by the Allis-Chalmers Manufacturing Co. is shown in Figs. 109, 110 and 110*a*. This type is built in single- and twin-tandem units and is designed particularly for blast-furnace gas.

The frame is of the box girder section and carries only one bearing as the engine is of the side-crank type. The material is cast iron and many cross webs are used to give stiffness.

The cylinders are each cast in one piece which practice is in direct contrast to the practice of the Mesta Machine Co. A

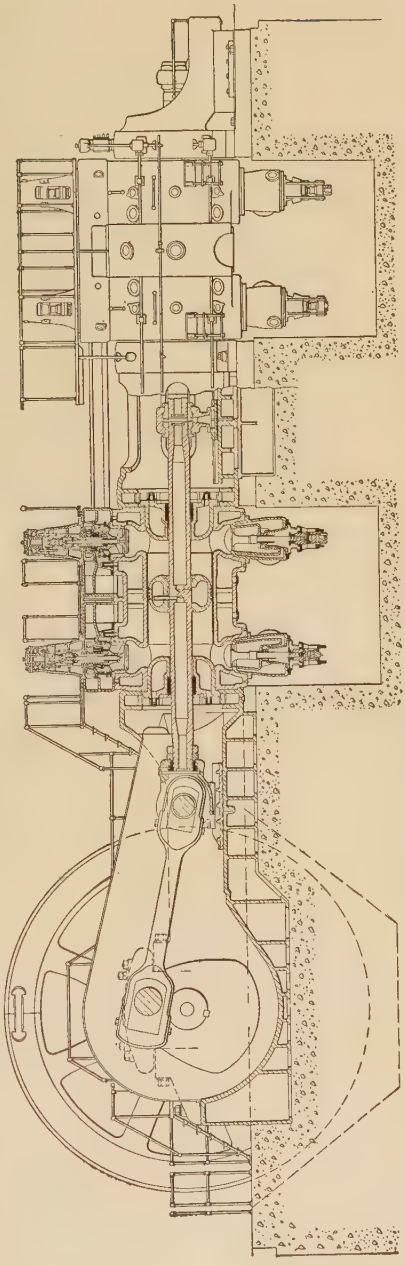


FIG. 109.—Allis-Chalmers Gas Engine Section and Elevation.

thin liner of cast-iron is used to take the wear. The main cylinder casting is of steel. The water-jacket is cast continuous from end to end. It will be noticed that the cylinders have no bearing on the foundation but are supported entirely by the frames and distance-piece.

The pistons, which are of cast iron, are very simple in design and cast with generous curves to prevent cracking due to expansion and contraction resulting from changes of temperature. They are fastened to the rods by being forced against a shoulder by a nut which is afterwards turned off flush with the face of the piston. Four cast-iron rings are used.

The piston-rods are large in diameter with large holes through them except at the middle where a wall is left intact to assist in forcing the cooling water through the piston. The front and back rods are duplicates.

The crossheads are all of the slipper type having only one slide. The main crosshead is forked and vertical adjustment is secured by means of shims. The usual practice of having the intermediate crosshead act as a coupling for the two parts of the piston-rod is followed in this case.

The connecting-rod is an open-hearth steel forging and is made solid at both ends. Adjustments for wear are made by means of wedges operated by screws. Adjustment at the crosshead end lengthens and at the crank-pin end shortens the rod. Babbitt-lined shells are used for bearings.

The main-shaft is of the built-up type and the crank disc is cast integral with the counter-weight.

Valve-gear.—The **cam-shaft** is driven direct from the main-shaft by means of helical gears. The main inlet- and exhaust-valves are operated by eccentrics, there being one eccentric at each end of each cylinder. The **exhaust-valve** is of cast-iron and is cast hollow to facilitate water-cooling. The disc of the main inlet-valve is also of cast-iron and is screwed onto the stem. Double springs are used with both of these valves. Both of the main valves receive their motion from the eccentric through rolling levers and these motions are fixed. In other words, the timing of the main inlet-valve is not changed to secure governing effect.

Referring to Fig. 110*a* it will be noticed that in addition to the main inlet-valve there are two auxiliary sleeve-valves, one con-

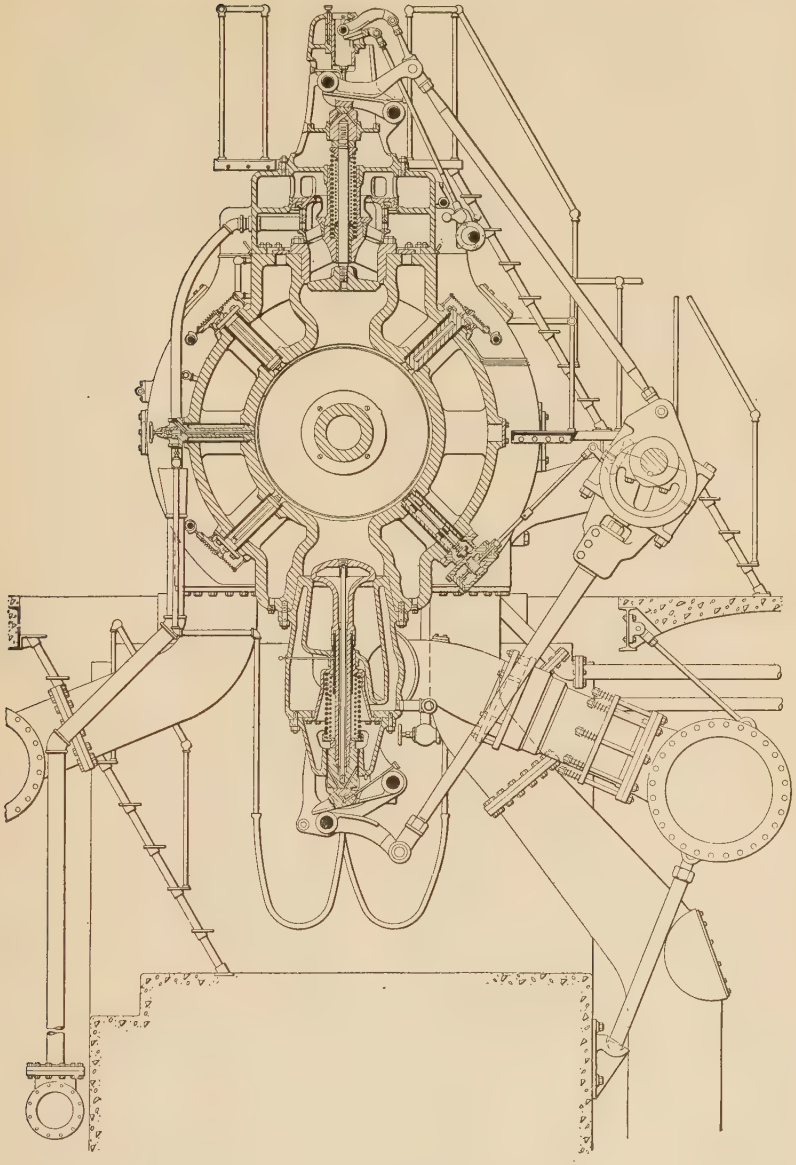


FIG. 110.—Allis-Chalmers Gas Engine. Section through Combustion Chamber.

trolling the gas and the other the air. These valves are operated by two rolling levers, one of which receives its motion from the main-valve push-rod, and the other from a separate governor-shaft. The governor-shaft is under the control of the governor

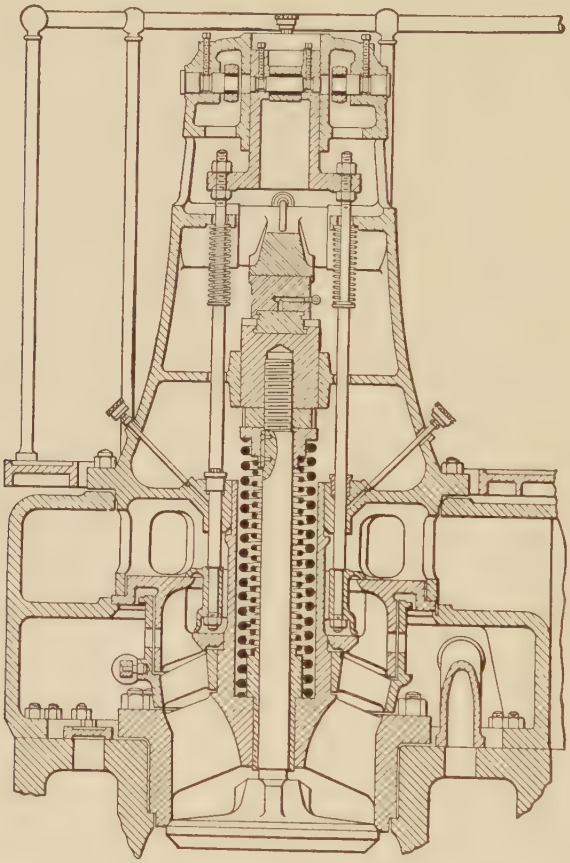


Fig. 110*a*.—Allis-Chalmers Gas Engine. Detail Inlet Valve and Governor.

and in this way the auxiliary sleeve-valve openings are controlled by the governor which varies the time of closing and thus regulates the amount of mixture drawn in to suit the load. At very light loads the amount of air per unit of gas is increased in order to hold up the compression pressure.

Ignition is accomplished by means of a make-and-break system, there being three igniters for each end of each cylinder. These igniters are shown in Fig. 110.

The air starting-valve is shown in section in Fig. 110. This is a double-beat poppet valve operated from an eccentric on the cam-shaft through a bell-crank and a push-rod. Between this poppet distributing valve and the cylinder is a check valve which closes automatically when the engine begins to fire.

CHAPTER XV

DIESEL ENGINES

Introduction.—About 1892 Rudolf Diesel published a study of possible internal combustion engine cycles, and decided that the best working cycle would possess the following characteristics:

1. Production of the highest temperature solely by compression.

2. Introduction of the finely divided fuel at such a rate that there would be no temperature rise during combustion.

3. Use of a large volume of air in proportion to the weight of the fuel to make it easy to keep down the mean temperature.

The first and second conditions were modified in the earliest successful practice to approximately constant pressure combustion with rise in temperature. The Brayton engine, as built 20 years earlier, had already used this modified cycle.

While the Diesel engine thus substantially follows the Brayton cycle, the mode of operation is different from that of the Brayton engine. Diesel did not begin to inject the fuel into the air until the end of the compression stroke. Brayton compressed the mixture of oil vapor and air together.

The Diesel engine of the present day is commonly described as an internal combustion engine in which liquid fuel is injected in a finely divided condition into highly compressed air at the end of the compression stroke, and at such a rate that there is no pressure rise during the spontaneous combustion that follows.

Diesel engines may work on either the four-stroke cycle or on the two-stroke cycle.

In the Diesel engine, the temperature at the end of the compression stroke must be high enough to cause spontaneous ignition of the pulverized fuel which is injected.

Two methods are in use for injecting the liquid fuel. The older and more common method is to blow in the oil as a fine mist by the use of highly compressed air. Another method now quite extensively used is to force in the oil, unmixed with air, through a very small orifice under a pressure of several thousand

pounds per square inch. The latter method is variously termed mechanical injection, or solid injection, or airless injection.

With mechanical injection, a compression pressure of 350 pounds per square inch will be high enough to cause spontaneous ignition. With air injection, from 450 to 550 pounds per square inch is used.

Where mechanical injection is used there is often a rapid pressure rise during combustion. This pressure rise may be so great that combustion is substantially complete at constant volume. Such engines are sometimes called explosion oil-engines. The engine then works on the Otto cycle, or constant volume combustion cycle, and has the same ideal efficiency as the Otto cycle for the same compression.

With mechanical injection the rate of fuel feed may be such that part of the combustion proceeds at constant volume, and the remainder at approximately constant pressure.

If ignition is solely due to the heat of compression of the air, engines of the two latter types are properly called Diesel engines, and were covered by a claim in the original Diesel patent.

Combustion at constant pressure, combustion at constant volume, and a combination of the two, with injection at or near the end of the compression stroke, are also used in connection with moderate compressions, from 100 to 250 pounds per square inch. With such compression pressures the air is not hot enough to cause spontaneous ignition of the fuel injected, and ignition and combustion are made certain by keeping a small part of the surface of the combustion chamber at a temperature of incandescence. The fuel is sprayed against this hot surface or brought into contact with it by air currents and the oil is evaporated and ignited by the hot spot or hot bulb.

Such engines are not called Diesel engines, but are termed surface-combustion engines, or semi-Diesel engines, or high-pressure oil engines. These engines have lower mean pressures and lower efficiencies than the Diesel engine.

Still lower compressions down to 60 pounds per square inch are used with hot bulbs or hot plates. In these engines the oil is injected early in the compression stroke.

In the Diesel engine the charge is composed of the following constituents; the main air supply, which is compressed in the power cylinder, the injection air supply which is compressed in

an auxiliary compressor, and the oil which is discharged by the fuel pump.

In the cycle diagram it will be convenient to consider the separate processes as occurring in a single cylinder. The equations will be somewhat simpler and just as general if we consider the action of just one pound of the fuel mixture.

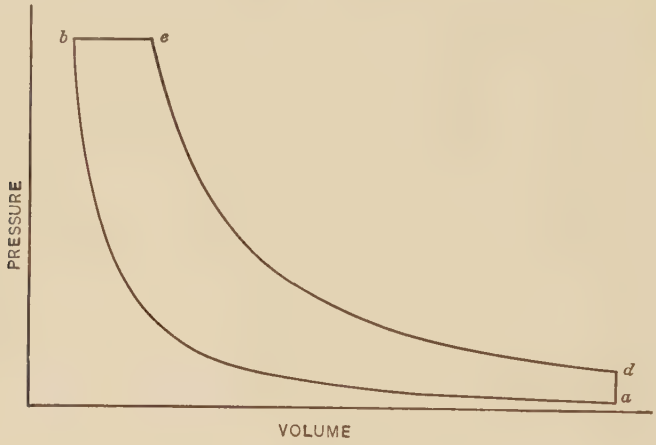


FIG. 111.

Call C_p and C_v the specific heats of the mixture at constant pressure and at constant volume, and n their ratio; also call Q_1 and Q_2 respectively the heat received during combustion and the heat rejected during exhaust, and finally call r the ratio of total volume to clearance volume and e the ratio of "cut-off" volume (or volume at end of combustion) to the clearance volume.

Let a, b, c, d, Fig. 111, be a pressure-volume diagram of the Diesel cycle with constant pressure combustion, with adiabatic compression and expansion. The efficiency will be:

$$E = 1 - \frac{Q_2}{Q_1} = 1 - \frac{C_v(T_d - T_a)}{C_p(T_c - T_b)} \tag{1}$$

Substitute $n = \frac{C_p}{C_v}$ and factor out the compression-line temperatures. This gives:

$$E = 1 - \frac{T_a\left(\frac{T_d}{T_a} - 1\right)}{nT_b\left(\frac{T_c}{T_b} - 1\right)} \tag{2}$$

But, by Charles' law, $\frac{T_c}{T_b} = \frac{V_c}{V_b} = e$

Also, $\frac{T_b}{T_a} = \left(\frac{V_a}{V_b}\right)^{n-1} = r^{n-1}$ for adiabatic compression

Also $\frac{T_d}{T_c} = \left(\frac{V_c}{V_d}\right)^{n-1} = \left(\frac{V_c}{V_b}\cdot\frac{V_b}{V_d}\right)^{n-1} = \left(\frac{e}{r}\right)^{n-1}$

And $\frac{T_d}{T_a} = \frac{T_d}{T_c}\cdot\frac{T_c}{T_b}\cdot\frac{T_b}{T_a} = \left(\frac{e}{r}\right)^{n-1} e.r^{n-1} = e^n$

Substituting in Eq. (2) we get

$$E = 1 - \frac{1}{nr^{n-1}}\,\frac{(e^n - 1)}{(e - 1)} \tag{3}$$

when $e = 1$, this reduces to:

$$E = 1 - \frac{1}{r^{n-1}} \tag{4}$$

This is the same as the efficiency of the Otto cycle for the same compression ratio. For zero load, $e = 1$; so that, for zero load, the ideal efficiency of the Diesel cycle equals that of the Otto cycle for the same compression ratio.

For greater loads e is more than unity, and the efficiency falls off. If we assume 32 pounds of air drawn into power cylinder per pound of oil burned, which is a ratio met in practice, at full load, and further assume 3 pounds of injection air and a mean specific heat of 0.26, the temperature rise during combustion will be

$$\frac{18{,}500}{(32 + 3 + 1)0.26} = 1980° \text{ F.}$$

If the suction pressure is assumed 13.7 pounds and the compression pressure 500 pounds, the compression pressure ratio is $\frac{500}{13.7} = 36.5$. Assuming $n = 1.32$, the volume ratio will be $36.5^{\frac{1}{1.32}} = 15.2$ and the clearance in per cent of stroke volume $\frac{100}{15.2 - 1} = 7.04$. The temperature ratio is $36.5^{\frac{1.32-1}{1.32}} = 2.395$. For an initial temperature of 120° F., the compression temperature becomes $2.395(460 + 120) = 1390$ degrees absolute, or 930° F.

The calculated combustion temperature is $1980 + 930 = 2910°$ F., and the value of e, assuming Charles' law to apply is $\frac{2910 + 460}{930 + 460} = \frac{3370}{1390} = 2.425$. This permits us to calculate the ideal efficiencies at various actual loads.

A usual assumption is that friction horse-power is constant. On this basis we may calculate the following table, for a four-cycle engine with 80 per cent mechanical efficiency at full load.

Actual load, per cent	Indicated power, per cent	e	Ideal indicated thermal efficiency	Expected actual conditions	
				Indicated thermal efficiency n = 1.40	M.e.p.
125	150	2.710	0.573	0.44	109
100	125	2.425	0.584	0.45	91
75	100	2.140	0.600	0.46	73
50	75	1.855	0.615	0.47	55
25	50	1.570	0.629	0.48	36
	0	1.000	0.663	0.51	0

Here

Efficiencies.—Actual four-stroke cycle engines of the largest sizes with more than 100 per cent excess air give the mean effective pressures shown in table and approach the tabular values for indicated thermal efficiency.

The indicated thermal efficiency of two-stroke cycle and four-stroke cycle engines should be about the same. The mean effective pressure of the two-stroke cycle engine may be a small amount greater where high scavenging pressures are used and perfect scavenging attained. The mechanical efficiency of the two-stroke cycle engine is necessarily lower due to the power required to operate the scavenging pumps.

The indicated power of a four-stroke cycle Diesel engine at full load is utilized as follows: Useful work 75 to 80 per cent, air compressor 10 per cent, other losses 15 per cent.

For the two-stroke cycle Diesel engine, this distribution becomes: Useful work 65 to 70 per cent, air compressor 10 per cent, scavenging 10 per cent, other losses 15 per cent.

In both cases, with careful design, higher efficiencies may be obtained, but useful work seldom exceeds 80 per cent for the four-stroke cycle engine and 70 per cent for the two-stroke cycle engine. These high efficiencies may be expected for large slow-speed engines.

The losses are about the same in total amount at all loads. This causes a rapid drop in mechanical efficiency at light loads. Calling the full load indicated thermal efficiency equal 45 per cent and the mechanical efficiency at full load equal 80 per cent for a four-stroke cycle Diesel engine, the expected efficiencies at light loads will be as follows:

Load, per cent	100	90	80	70	60	50	40	30
Indicated thermal efficiency	0.45	0.46	0.46	0.46	0.47	0.47	0.48	0.48
Mechanical efficiency	0.80	0.78	0.76	0.73	0.70	0.66	0.61	0.54
Thermal efficiency on delivered horse-power	0.36	0.36	0.35	0.34	0.33	0.31	0.29	0.26
Pounds oil of 18,000 B.t.u. per pound per delivered horse-power per hour	0.39	0.39	0.41	0.42	0.43	0.46	0.49	0.55

From half load to full load these figures correspond fairly well with the best present day economies reached with four-stroke cycle Diesel engines, using Diesel engine oil.

Manufacturers' guarantees of fuel consumption are about 10 per cent higher from full load to half load for the heavy slow-speed type. For the lowest loads given in the table the proportion of the indicated thermal efficiency to the ideal efficiency is lower. Delayed combustion and incomplete combustion, especially with poor oils may raise the rate of fuel consumption above the figures given for the lowest load.

For two-stroke cycle engines with the mechanical efficiency equal 70 per cent, and the thermal efficiency equal 45 per cent at full load, the following table gives a reasonable estimate of thermal efficiency and fuel economy with Diesel oil of good low-speed engines at varying loads.

Load, per cent	100	90	80	70	60	50	40	30	20
Indicated thermal efficiency	0.45	0.46	0.47	0.47	0.48	0.48	0.49	0.50	0.50
Mechanical efficiency	0.70	0.67	0.65	0.62	0.58	0.53	0.48	0.41	0.31
Thermal efficiency based on delivered horse-power	0.31	0.30	0.30	0.29	0.27	0.25	0.23	0.20	0.15
Pounds oil of 18,000 B.t.u. per pound per delivered horse-power hour	0.46	0.47	0.47	0.49	0.53	0.57	0.62	0.71	0.95

Two-cycle engines of the largest size where the work of scavenging pump and air compressor are reduced, especially at light loads, will give considerably better thermal efficiencies and lower fuel consumption than those given in the table.

DIESEL ENGINES—DIMENSIONS

In Diesel engines the mean effective or average net pressure of the gas on the piston during a complete cycle is determined by the point of cut-off of fuel admission, by the thermal efficiency realized, and by the volumetric efficiency.

Two-stroke cycle Diesel engines, being provided with scavenging pumps, which deliver air under pressure, may have higher volumetric efficiencies than four-stroke cycle engines which have to draw in air at less than atmospheric pressure. The volumetric efficiency of slow-speed two-stroke cycle engines may be taken as 95 per cent while for four-stroke cycle engines the volumetric efficiency is generally a few per cent less.

The maximum mean effective pressure realizable from a two-stroke cycle Diesel engine will be therefore somewhat greater than that of four-stroke cycle engines.

However, the much higher average temperatures which necessarily prevail in the cylinders and heads of two-stroke cycle engines make the stresses encountered greater in this type for the same mean effective pressure. Accordingly, in practice, somewhat lower mean effective pressures are usual for the two-stroke cycle engine. The table below gives average values for the mean effective pressures usually developed at the rated load. All Diesel engines can develop at least 20 per cent overload capacity for short intervals, and therefore can develop higher mean effective pressures than shown in the table.

MEAN EFFECTIVE PRESSURES IN POUNDS PER SQUARE INCH AT RATED LOADS

	Four-stroke cycle	Two-stroke cycle
Slow-speed	95	90
High-speed	90	80

There may be a variation of about 5 per cent above or below these figures in the best types of Diesel engines.

Another common figure used in discussing Diesel engines is the product of the mean effective pressure by the mechanical efficiency.

Taking the mechanical efficiencies as 78 per cent and 72 per cent for slow-speed and high-speed four-stroke cycle engines, and 70 per cent and 65 per cent for the corresponding two-stroke cycle engines, we get the following table for what may be called the mean effective pressure *reduced* to the brake or shaft.

MECHANICAL EFFICIENCIES TIMES MEAN EFFECTIVE PRESSURES FOR DIESEL ENGINES

	Four-stroke cycle	Two-stroke cycle
Slow-speed	74	63
High-speed	65	52

Ratings greatly in excess of these figures can only be justified on the ground that every feature of the design is of the greatest excellence and that both the construction and materials are of the very best obtainable.

The piston speeds of Diesel engines may be averaged as follows:

	Four-stroke cycle	Two-stroke cycle
Slow-speed	700	800
High-speed	1000	800

There is a considerable variation, and speeds as low as 600 and as high as 1200 are in use. It is to be noted that the two-cycle high-speed, or high-revolution engine has not been extensively

used at the high piston speeds which are quite common with fast running four-stroke cycle engines.

The average ratio of stroke to diameter and the average revolutions per minute are given below.

	Ratio stroke to diameter	Revolutions per minute
Slow-speed four-stroke cycle.....	1.4	100 to 200
Slow-speed two-stroke cycle.....	1.6	100 to 200
High-speed four-stroke cycle.....	1.2	400 to 600
High-speed two-stroke cycle.....	1.3	400 to 600

The extremely high revolutions are confined to marine engine of small powers and small cylinder sizes.

The effective horse-power per cylinder for four-stroke cycle stationary engines does not often exceed 125 for slow speed and 150 for high speed. Two-stroke cycle engines are built in larger cylinder powers, 250 to 300 effective horse-power per cylinder being not uncommon.

FUELS

The fuel used in Diesel engines is generally a filtered fuel oil. The important requirements are: High flash point, above 125° F., freedom from dirt, ash, or water; not more than 2 to 3 per cent sulphur; low viscosity; specific gravity from 0.85 to 0.95; and of not less than 18,000 B.t.u. per pound. Generally oils high in hydrogen are best.

Fuel oils made from crude oils of both paraffine base and asphaltum base are used. The asphaltum base oil may require heating to make it flow freely.

Tars from lignite may be used without trouble.

Coal tar and tar oil are less satisfactory as fuels. The benzol or benzol derivatives, which appear in tar oils, are not easily decomposed, and accordingly ignition is difficult. Sometimes these oils are used in connection with a lighter "ignition oil."

Vegetable oils may be used. At present, because their price is so much higher than petroleum oils, they are not used.

Sulphur was formerly considered very objectionable because of its corrosive action. With the present system of lubrication, sulphur content up to 3 per cent is not objectionable.

Water is likely to cause failure of ignition. The mineral salts carried by the water may also cause incrustation and fouling of the atomizer parts.

Ash or coke will mix with the lubricating oil and cause serious wear. Deposits on the valves may also be objectionable.

High viscosity makes pumping and atomization difficult. Asphaltum content which is associated with high viscosity may also cause gumming of the fuel needle. These troubles are greatly relieved by heating the oil to 100° F. or more.

The low price of petroleum excludes the considerable use of tar oil in the United States. In Germany it has long been used successfully, and in future it may become an important Diesel fuel here.

David White, Chief Geologist of the United States Coast and Geodetic Survey, has estimated the oil-well reserves in the United States at 9 billion barrels with a present rate of consumption of half a billion barrels a year. At this rate our oil wells will be exhausted in less than 20 years. However, we are already importing 20 per cent of our oil. The world's oil-well reserves are enough to last nearly two centuries.

The United States has, in addition, great deposits of oil shales. To utilize these deposits will require in the future the building up of an enormous mining industry comparable in size with our present coal mining industry.

With the supply of coal tars from bituminous coals the supply of fuel for Diesel engines is secure for centuries. The gradual exhaustion of the oil wells will more and more operate to restrict the use of fuel oils to Diesel engines.

GENERAL

The Diesel engine is almost always single-acting. In smaller and moderate sizes the wrist-pin is generally placed in the piston which is lengthened to act as a cross-head. This is the trunk piston construction which shortens the engine in the direction of the cylinder bore.

As the pressure intensity on the piston head reaches 500 pounds per square inch, or more, during every cycle, the frame, cylinder, reciprocating parts, and crank-shaft are much heavier and stiffer against vibration in the Diesel engine than in the gas engine or steam engine where lower maximum pressures are used.

For vertical engines this consideration has led to the general adoption of the A-frame or box-frame construction. For large engines, especially of the marine type, custom has required that the wrist-pin be open for inspection. For such engines separate cross-heads are generally used with single-slides, leaving the front side of the wrist-pin bearing exposed.

In the usual type Diesel engine, atomization of the fuel and its distribution throughout the combustion chamber is secured by means of compressed air at from 700 to 1000 pounds per square inch. A characteristic feature of the Diesel engine therefore is the air compressor required for fuel injection. For the very high-pressure ratio of about 964 pounds absolute to about 13.8 pounds absolute, equal 70 to 1, three-stage compressors are commonly employed.

The amount of free air handled by the compressor is about 11 per cent or 12 per cent of that drawn into the power cylinders; or in the neighborhood of 3 pounds of injection air per pound of fuel. The air is thoroughly cooled after each stage compression. With such a compressor the ideal mean effective pressure per stage would be about 19 lb. per sq. in., or $3 \times 19 = 57$ lb. per sq. in. total, based on volume of air drawn in. The ratio of the indicated work in the compressor to indicated work in the power cylinder is, accordingly, for 91 pounds mean effective pressure in power cylinder, $\frac{57 \times 0.115}{91} = 0.074$.

In the two-stroke cycle engine a scavenging cylinder is also required. The amount of scavenging air is commonly 1.3 to 2.0 times the total displacement of the power cylinders. The discharge pressure is from 3 to 7 pounds gauge. The indicated work will be about the same as that for the air compressor.

The fuel valve and fuel pump are also distinctive features. The lubrication, governing, starting, and reversing mechanisms correspond more closely with the corresponding parts of the gas-engine.

CONSTRUCTION OF FOUR-STROKE CYCLE DIESEL ENGINE

Fig. 112 shows a side elevation, partly in section, of a $13\frac{1}{2}$ by $19\frac{1}{2}$ by 225 r.p.m., 172 brake horse-power, four-stroke cycle, Fulton-Diesel engine manufactured by the Fulton Iron Works of St. Louis, Mo.

This engine has three power cylinders. The cylinder to the extreme left is the air compressor. Close inspection will show that the air compressor piston has three diameters. The smallest diameter is above, then the largest diameter, then the intermediate diameter.

On the down stroke air is drawn in through the silencer and control valve which projects to the right. The air then enters the inclined plate valve shown opening into the space between the

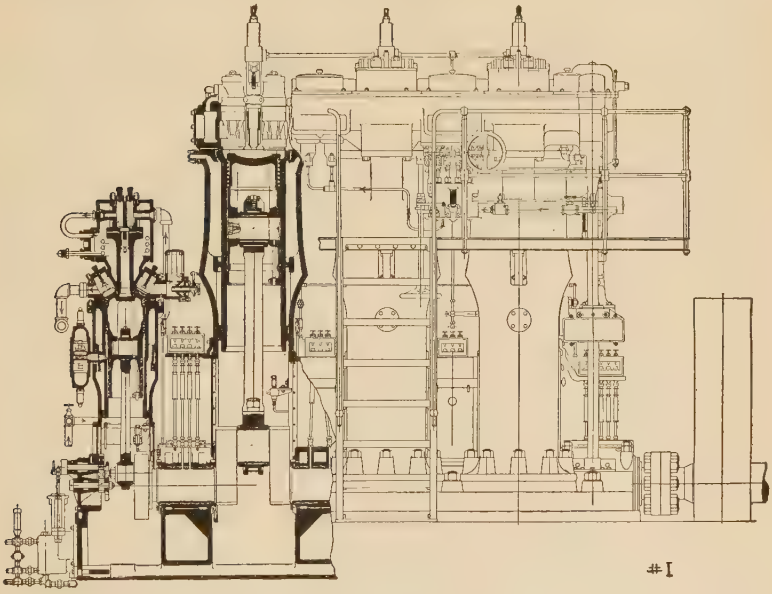


FIG. 112.—Three-cylinder Fulton Diesel Engine. Section and Elevation.

largest and the smallest piston diameters. Near the end of the up-stroke this air is discharged at a pressure of about 60 pounds through the inclined plate valve to the left.

This air is passed through a cooling coil, not shown in this view, and, on the next stroke, enters through a plate valve, shown to the left and lower down, to a space swept by the difference in diameters between largest and intermediate pistons. On the return stroke this air is discharged through the plate valve in the same cage and passes through another cooling coil, not shown. On the next stroke the air is drawn in through the right hand

poppet valve indicated at the extreme top of the compressor and enters the space above the smallest piston. On the return stroke the highly compressed air is discharged through the other poppet valve and passes through an after cooler composed of the pipe coil shown surrounding the upper part of the compressor cylinder. By dividing the compression into three stages with intercoolers between stages, the work of compression is reduced, very small clearances are avoided, and, perhaps most important of all, the temperature rise in each stage is kept down to 250 or 260° F.

The lubricating oil pump will be seen at the extreme left, being driven by a return crank and a pair of gears from the compressor crank-pin.

Just to the right of the compressor will be seen one of the power cylinders in section. Note the long trunk piston with the thick piston-head, and the very large piston-pin and crank-pin. The oiling device for the piston-pin is shown to the right of the connecting-rod head. An oil guard across the piston is shown above the wrist-pin bearing.

In the cylinder head above are shown the injection valve with the inlet and exhaust valves on either side.

Above the platform and between the other two power cylinders will be seen the starting hand wheel. This wheel, through suitable shaft and cams, controls the operation of the fuel pumps to the several cylinders and the admission of compressed air to the automatic starting valve. Rotation of the hand wheel to the first notch admits starting air to one or more cylinders, leaving fuel pumps idle. After the engine makes a few revolutions the wheel is rotated to the second notch, which brings into operation the fuel pumps of those cylinders not operating on air. As soon as these cylinders begin to develop power the wheel is advanced to the third notch or running position. This shuts off the starting air and brings all of the power cylinders into operation on oil. The hand wheel shown behind the stairway may also be used for starting.

The vertical shaft to the extreme right operates the governor shown about at the middle and is geared to the horizontal half-time shaft or valve-operating shaft at the top of the engine.

Several sight feed oilers with three to five feeds are used for oiling the several main bearings, crank-pins and wrist-pins.

Fig. 113 shows an end view from the compressor side. The circulation of air to and from the compressor cylinders can be followed by arrows. The water cooled exhaust pipe and a silencer for air inlet to power cylinder can be seen to the left. To the right is seen the operating platform, the fuel pumps and

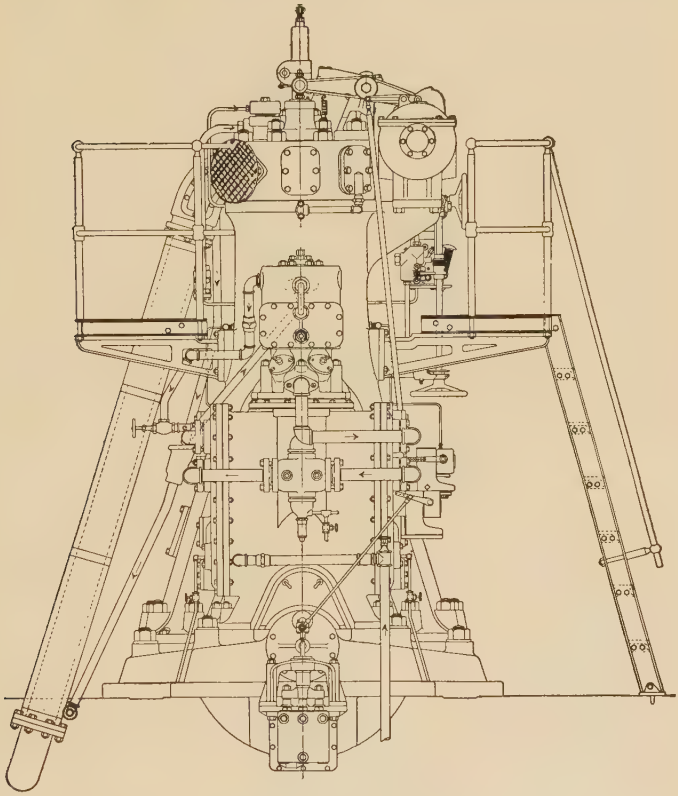


Fig. 113.—Compressor End of Engine Shown in Fig. 112.

the two starting hand wheels. The levers at top are for operation of inlet, exhaust and fuel valves. To the left of these three valves is seen the upper end of the air starting valve housing.

Fig. 114 is a sectional end elevation through a power cylinder. Important points to be noted are: The cylinder oiling from both sides entering below the second piston ring with the piston all the way down, the method of securing the liner in the cylinder;

the half-time shaft or cam-shaft; the fuel inlet or injection valve operation; with the method shown for altering its lift, by the operator; and the two air pipes to the air starting valve, the upper one for timing and the lower one for supplying the starting air.

This engine governs by varying the stroke of the fuel pump. Sizes from 100 to 600 brake horse-power are built.

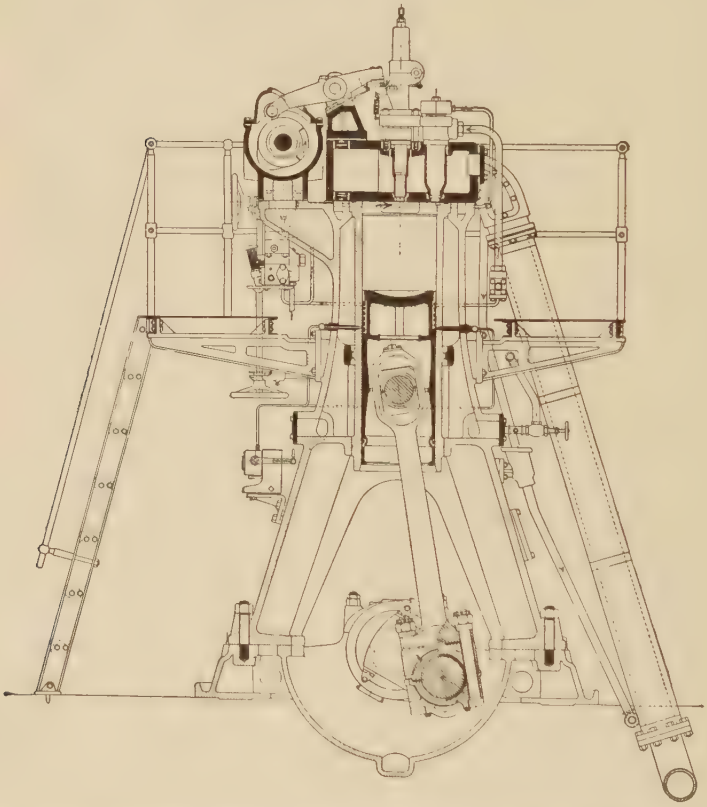


Fig. 114. –Transverse Section through Power Cylinder, Engine Shown in Figs. 112 and 113.

The Busch-Sulzer Bros. Diesel Engine Co. of St. Louis, Mo., builds both four-stroke cycle and two-stroke cycle Diesel engines. This company builds the four-stroke type in sizes up to 500 brake horse-power and the two-stroke type in the larger sizes, especially for marine use.

TWO-STROKE CYCLE ENGINES

Fig. 115 shows a sectional end elevation for Type C, two-cycle Busch-Sulzer Diesel engine. This engine is of the single-acting,

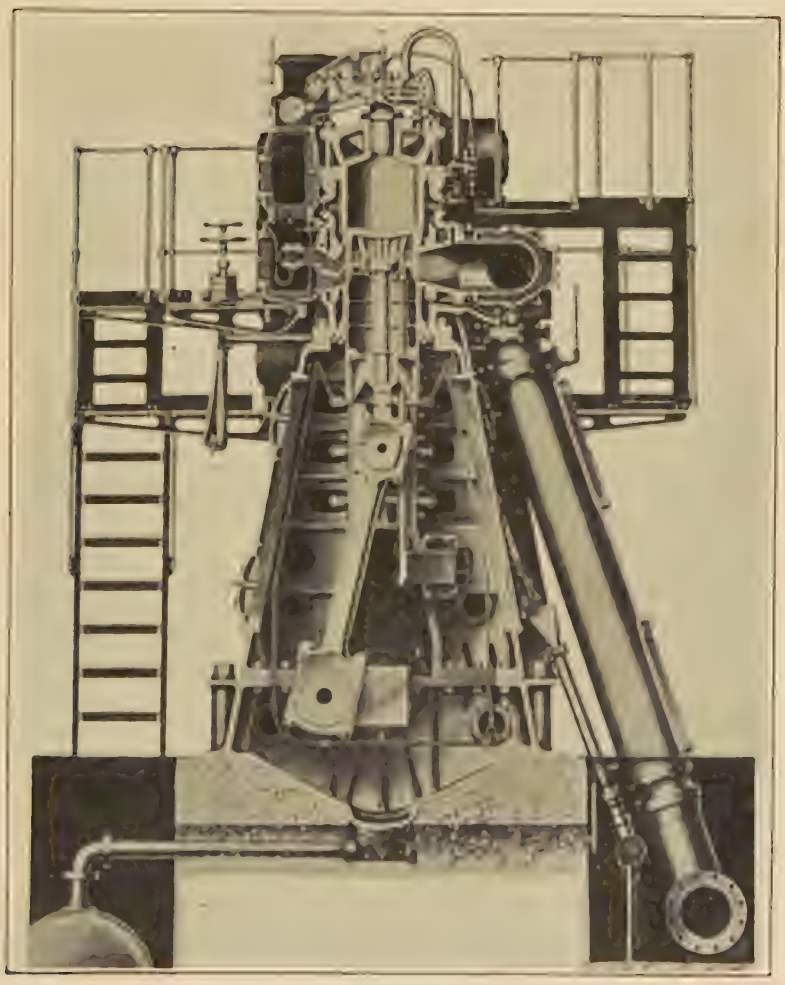


Fig. 115.—Busch-Sulzer Bros. Diesel Engine.

cross-head type, with water-cooled piston. The scavenging and exhaust ports are clearly seen, but will be described more fully later. The extremely simple cylinder head is a most excellent

feature. The valve housing in the center of the head contains the fuel inlet valve and the air starting valve. The bed plate and the crank-case are of very strong, rigid construction.

Fig. 116 shows a general view of a 1100 brake horse-power, four-cylinder, Busch-Sulzer two-cycle marine engine. The scavenging cylinders are placed in line with the power cylinders and are driven by a separate crank. In line with the power and scavenging cylinders, at one end of the engine is the air com-

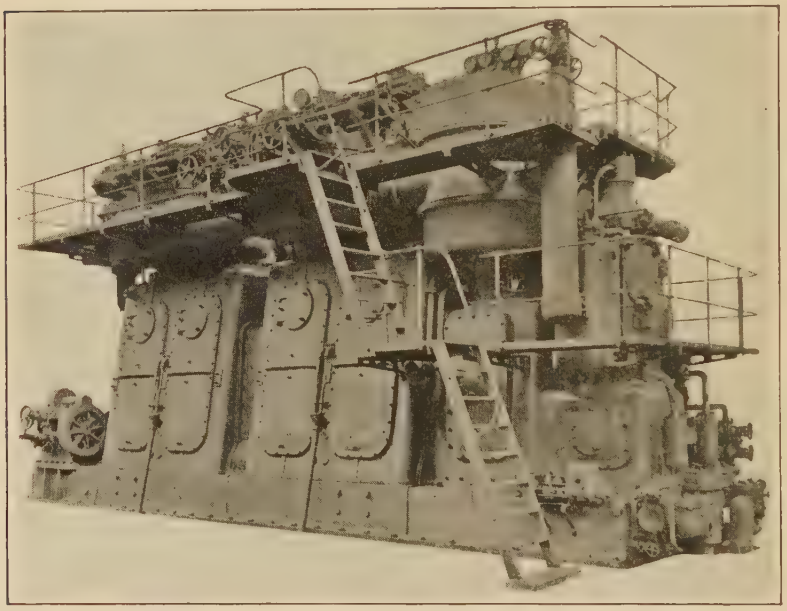


Fig. 116.—Busch-Sulzer Bros. Diesel Marine Engine.

pressor, also driven directly from a separate crank on the crank-shaft.

Fig. 117 is a vertical section through the power cylinder, showing scavenging and exhaust ports, piston and cylinder head.

The outer cylinder jacket is bolted to the crank-case below and to the cylinder head above. It sustains all the axial forces due to the pressure of compression and combustion. The inner cylinder, or liner, in which the piston reciprocates, is held between cylinder jacket and cylinder head. A single row of ports on one side of the liner provides for the exit of the burnt gases and a

double row of ports on the opposite side provides for the admission of scavenging and charging air. Packings between liner and jacket above and below the ports permit free expansion of the liner and prevent leakage of water or gas.

The lower tier of air ports is always in communication with the scavenging receiver where air at about 5 pounds per square

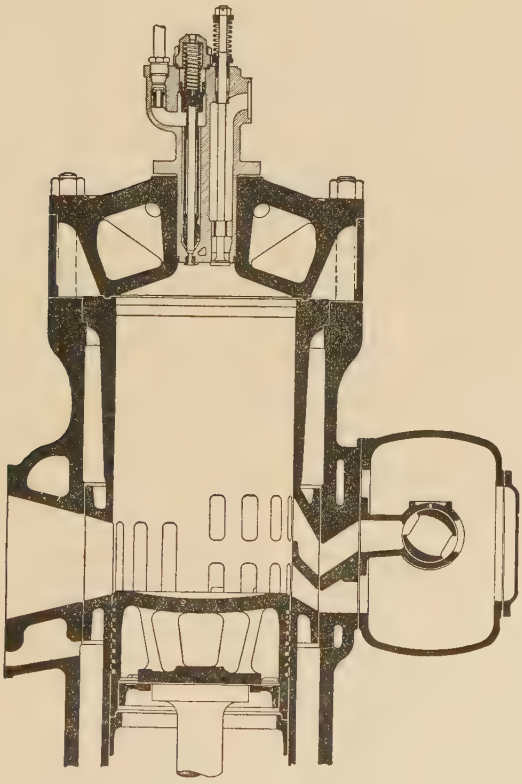


Fig. 117.—Section through Power Cylinder, Busch-Sulzer Bros. Diesel Marine Engine.

inch gauge pressure is stored. The upper tier of ports communicates with the same air receiver only when the timed rotary scavenging valve shown, is open.

On the down stroke of the piston, always a power stroke, the gases gradually expand from the compression and combustion

pressure of about 500 pounds. When the upper tier of scavenging ports is uncovered, the rotary scavenging valve is closed, and only a slight drop in pressure occurs, due to the sudden expansion of the gases into the ports leading to the rotary valve. A little later, about 10 per cent before the end of the stroke, the exhaust ports are uncovered by the descending piston and the gas pressure rapidly drops from 40 pounds to about atmospheric pressure during 8 per cent of the piston stroke. The lower tier of scavenging ports, always open to the receiver, is then uncovered and a strong flow of fresh air at 125° F. or less and driven by a moderate pressure of 5 pounds, or less, sweeps the burnt gases out of the cylinder. Just before the end of the stroke, the rotary scavenging valve opens admitting air through the upper row of ports, and, with the piston returning, greatly increases the efficiency of the process of cleaning out the burnt gas over that obtainable with only the lower tier of air ports.

The process of scavenging continues after the lower tier of scavenging ports are closed and until the exhaust ports are covered. During the next 8 or 10 per cent of the piston stroke, or until the upper tier of air ports is closed, the power cylinder is in communication with the scavenging receiver and the pressure is built up to about 5 pounds per square inch above atmospheric.

This arrangement of the double tier of air ports, with timed rotary scavenging valve, is covered by patents.

The beautifully simple construction of the cylinder head makes for a uniform distribution of the expansion due to heat flow, largely eliminates local temperature stresses, and permits operation at comparatively high speeds.

Fig. 118 is a vertical section through the scavenging pump. Two double acting pistons mounted in tandem on the same rod provide two discharges every revolution. A cross-head is necessarily used. No piston rings are required on account of the very light discharge pressure. Suction and discharge valves are identical and therefore interchangeable. They are of the automatic shutter type mounted in cages containing from half a dozen to a dozen and a half valves.

The air compressor is of the three-stage differential piston type with shutter valves in the first stage, ring plate valve in the second stage and mushroom valves in the third stage. Intercooling and aftercooling coils are provided.

The elimination of exhaust valves and the simplification of cylinder head design together with the double frequency of the power impulse makes the two-stroke cycle type especially well suited for large cylinder bores and great powers. For marine work the greater simplicity of the reversing gear is also an

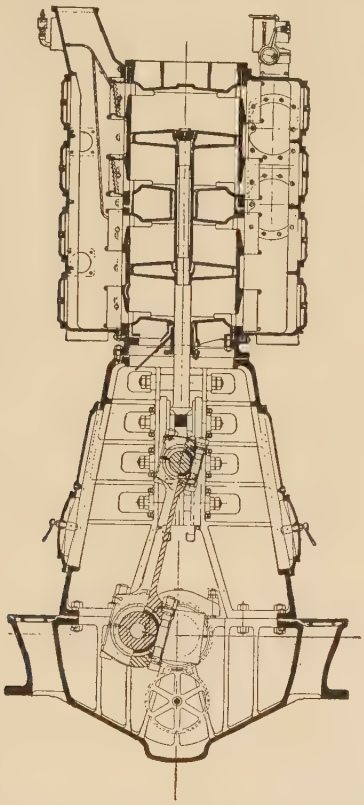


Fig. 118.—Section through Scavenging Pump. Busch-Sulzer Bros. Diesel Engine.

advantage. Against these advantages must be placed the increased fuel consumption, especially at the lighter loads.

At the present time the four-stroke cycle engine is probably ahead in marine service both in number of installations in operation and under construction, principally due to the pioneer work of one very large manufacturer of four-stroke cycle engines whose marine engines have been very successful.

The period of fuel injection is necessarily short, about one-twelfth of a revolution, which, at a speed of 120 r.p.m., corresponds to about four-hundredths (0.04) of a second. The earlier part of this brief period is devoted to ignition of the first particles of cold oil and the formation of a high temperature flame in the cylinder. In the few hundredths of a second remaining as much as a quarter of an ounce of oil must be spread through the air and burned, in a cylinder of 24 inches bore.

The oil must be carried to all parts of the combustion chamber and must be divided into particles so small that in the temperature of the surrounding air evaporation and combustion can be 100 per cent complete soon after fuel admission ceases.

The fuel particles cannot move in streams of uniform density and must make up for this lack of uniformity by having a velocity high enough to circulate many times about the combustion chamber in about one-hundredth of a second.

In the early history of the Diesel engine compressed air was found to be a satisfactory agent for atomizing and distributing the oil as it entered the combustion chamber. In this method still in general use, oil is pumped into a cavity containing air at 900 lb. or 1000 lb. pressure and located close to the combustion chamber. At the proper time, just before the beginning of the power stroke, a needle valve opened, communication with the power cylinder and the compressed air atomized the oil and drove it in through a small opening or nozzle.

Two somewhat different methods of atomization are used. In the first, the oil is pumped in on a number of perforated discs. On opening the fuel valve the compressed air passes through the holes in these discs driving some of the oil ahead and atomizing all of it quite thoroughly.

In the second method the compressed air by injector action draws up the oil from a recess and then atomizes it in the same manner that oil is atomized in nasal atomizer or a fruit tree sprayer.

Fig. 119 gives a diagrammatic view of the first type. The fuel needle is held tightly against the seat shown by the action of a spring. The oil from the fuel pump enters through the passage in the valve cage while the needle is closed. Air from the receiver and at 900 pounds pressure, or more, is admitted through the air passage shown, which is always open. On lifting the needle

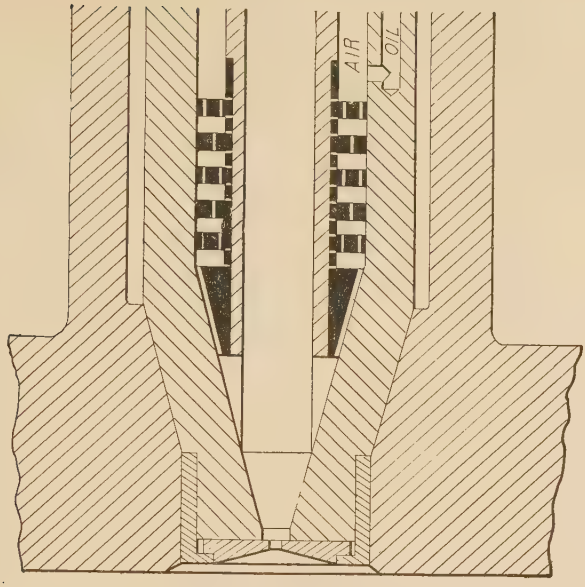


FIG. 119.—Fuel Injection Valve Type I.

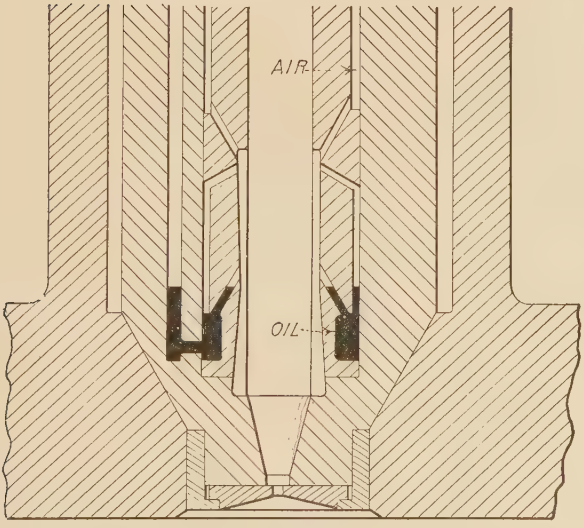


FIG. 120.—Fuel Injection Valve Type II.

some oil unmixed with air is necessarily blown in first and ignites. The air quickly makes an opening through the oil and carries in the remainder by friction. By regulating the number of discs and the size of the holes in the discs the atomization and mixing can be regulated.

Fig. 120 shows the injector type, sometimes called the Hesselmann fuel valve. Oil is pumped into the darkly shaded passages with the needle closed. The full injection pressure bears on both sides of this bent oil column. When the needle opens the escape of air causes a large drop in pressure at the point where the oil passages open into the narrow space around the needle. The difference in pressure on the two legs of each oil column causes the oil to flow out rapidly into the air stream and become thoroughly atomized.

A third type of nozzle, now quite extensively used on horizontal Diesel engines, admits oil into the passage below or beyond the needle. The needle protects the oil from the injection air pressure, and by pumping the oil at a time when there is little pressure in the power cylinder, the fuel pump is relieved from the necessity of operating against a high pressure. This makes governing easier.

Solid injection, originally used with surface ignition engines has come into use by a number of manufacturers of Diesel engines. The fuel pump delivers the oil through a collapsible tube and positive valve, or through an automatic valve, and then through one or more very fine orifices into the combustion chamber at the beginning of the power stroke. To secure good atomization, pressures of 2000 pounds to 4000 pounds per square inch have been used, and the size of the injection holes is as small as 0.02 inch in diameter. A 100 b.h.p. Vickers engine is quoted as having a five-hole nozzle with a total cross-sectional area of 0.0013 square inch.

Such an engine would use at least 40 pounds of oil per hour, the injection period being 30 degrees or one twenty-fourth the time of a cycle. The average rate of injection, then, would be about $40 \times 24 = 960$ pounds per hour or $\frac{960}{3600} = 0.267$ pound per second. For oil of $7\frac{1}{2}$ pounds per gallon this is equal $\frac{0.267}{7.5} \times 231 = 8.2$ cubic inches per second. The jet velocity will be, then:

Velocity $= \frac{8.2}{0.0013 \times 12} = 526$ foot-seconds. This is comparable to the velocity of projection of mortar shells from guns. The pressure required to furnish such a velocity is $P = \frac{12\,WV^2}{2g} = \frac{12 \times 0.0325 \times \overline{526}^2}{2 \times 32.16} = 1680$ pounds per square inch. Friction losses will increase this to more than 2000 pounds per square inch.

Air at 950 pounds per square inch, used for injection, expands to the compression pressure of 500 pounds by a series of pressure drops through the various discs. If the expansion were in a single pressure drop the air velocity would be about 1000 feet per second. With a ratio of 3 pounds air per pound oil, the mean velocity of blast would be $1000\sqrt{0.75} = 870$ foot-seconds. This is not very greatly in excess of the velocity of the oil jet in solid injection, and, as the air is strongly throttled in passing through the nozzle discs, it is probable that the actual jet velocity with solid injection is higher than with air injection.

The maximum drop below initial temperature with air injection is about 90° F. In practice, throttling reduces this greatly, so that the cooling of the air blast by expansion, so often spoken, about, is almost negligible with the usual relation of injection pressure to compression pressure.

The considerable weight of injection air used lowers the average temperature in the cylinder about 10 per cent. If air injection pressures used in starting are excessive, the increased weight of the blast, with its lower temperature, may dangerously lower the temperature in the cylinder.

With air injection, the weight of the injection jet is about four times as great as with solid injection. Therefore it is not so quickly slowed down by friction. For this reason, it is claimed that the mixing of fuel with the main air supply is more perfect with air injection than with solid injection.

The compressor must be of superior design and construction to handle pressures of 1000 pounds per square inch. Inferior compressors may be both unreliable and dangerous.

Multi-stage compression is essential to save the work of compression and to reduce or eliminate danger of explosion. Except with some open nozzle engines, the high-pressure air is always stored in a receiver. Therefore the air is cool when used.

Three-stage compressors are used except in the smallest sizes. Even in very small engines, three-stage compression would be much better, but the tiny valves required in the third stage, and the increased complication of a three-stage compressor have given the two-stage compressor a place in some engines.

For a total pressure ratio of 80, the pressure ratio in each cylinder is 4.32 with three-stage compression and 8.95 with two-stage compression. The temperature ratios, for the compression exponent equal 1.3, is, for the three-stage compressor 1.4, while for the two-stage compressor it is 1.66. If air enters at 100° F., the temperature rise will be 224 degrees for three-stage and 370 degrees for two-stage. With very efficient cooling and low speeds of operation, the compression exponent may be as low as 1.2 with a corresponding temperature rise of 157 degrees in a three-stage compressor and 246 degrees in a two-stage compressor.

Even with the best design the cooling may be expected to be imperfect at times due to local obstructions in water passages or due to scaling of cooling surfaces with a deposit from the cooling water. A $2\frac{1}{2}$ per cent clearance reduces the volumetric efficiency about 5 per cent in a three-stage compressor and about 12 per cent with two-stage compression.

After each stage compression, including the last, the air is passed through a cooling coil, and finally the oil and water is removed in a separator.

The valves in the low-pressure stage are usually feather or plate valves. In the high-pressure stage poppet or mushroom valves are used on account of their superiority for high-pressure work. A leak in a discharge valve is dangerous as the leakage is essentially a throttling operation, with little loss in temperature. The hot air just compressed will then mix with the charge, being compressed, raising its temperature and raising the maximum temperature at the end of the stroke. If the discharge valve leaks, it will leak most during the suction stroke. This displaces the entering charge, causing a serious loss in capacity and results in increased work and increased temperature in the stage below as well as in the stage where the leakage occurs.

The compressor usually furnishes the air for starting as well as for fuel injection. The capacity supplied for these two purposes varies from 10 to 15 per cent of the displacement of the

power cylinders. In marine installations a reserve compressor is usually supplied.

In two-cycle engines the scavenging cylinder often acts as the first stage of the compressor.

Compressors are most commonly vertical, and may be arranged on an extension of the crank-shaft or may be driven by levers from the connecting-rod. The latter construction saves length but is not so convenient for inspection.

The injection pressure should be reduced at light loads. This may be done by hand. Another way is to have the governor operate a damper or throttle valve in the suction of the compressor. This increases the pressure ratio and the maximum temperatures in all stages. A better way is to have a governor control arranged to hold open a suction valve on the low-pressure stage during part of the discharge stroke.

The air receivers are preferably built of seamless steel or copper. They are long and small in diameter. The capacity of the injection air receiver is enough to hold the entire discharge of the compressor at maximum pressure for about 4 minutes. This is about 0.02 cubic feet per brake horse-power of engine. Two other receivers from 5 to 10 times as large are provided for the starting air. The starting air receiver pressure is not allowed to fall much below 500 pounds per square inch. Pressures as low as 200 pounds may be used, however. Safety valves are always required.

SCAVENGING

Practice is not standardized in scavenging two-stroke cycle engines. Some manufacturers, following the uniflow idea have scavenging valves in the cylinder head which open after the piston over runs the exhaust ports. The air entering behind the column of burnt gas can drive the gas out with a minimum of power, of turbulence, and of mixing. That is, complete scavenging is more readily effected.

The other type two-cycle construction has a set of scavenging ports in the cylinder wall opposite the exhaust ports. The advantages of this construction are the larger area of the ports, the greater simplicity of the cylinder head, and the lower scavenging pressures associated with larger port area. The claim of greatly

superior efficiency made by the builders of the scavenging valve in the head type is disputed.

Scavenging pumps, like air compressors, are operated either from an extension to the crank-shaft, or through levers driven from the connecting-rod or cross-head. In some types where space is very limited stepped pistons are used, the smaller diameter for the power piston, the difference in the two diameters for scavenging.

Scavenging pistons are almost always double acting and generally two pistons are in tandem on the same rod. Pressures are 3 to 7 pounds per square inch, and piston rings may be dispensed with. Valves are now automatic and of the feather or plate type.

To reduce fluid friction, the pipes from scavenging pumps to power cylinders are made very large and act as receivers.

The displacement of scavenging pistons is from 1.3 to 2.0 times that of power pistons, the larger displacements being used in port scavenging engines. The ports are designed to pass the charge at a velocity of 300 feet per second, and ordinarily take up about 30 per cent of the stroke.

FUEL PUMPS

The fuel pump displacement is always several times the quantity of fuel oil required. A separate pump is usually employed for each cylinder of a multi-cylinder engine. A type is now built in which a single plunger makes as many strokes per cycle as there are power cylinders. For this single plunger there are as many discharge valves as cylinders. For each cylinder in turn one of the discharge valves, controlling a pipe line leading to that cylinder, opens at the proper time.

The general method of governing is by regulating the time of closing the pump suction valve. This requires a valve-operating eccentric as well as a plunger eccentric. Delayed closing causes the oil to be sent back to the suction line during more or less of the discharge stroke.

Hand plungers are sometimes provided for pumping oil to the fuel valves before starting.

Where the oil is fed to the pump by gravity from a tank higher than the injection valve, the fuel oil pipes are primed by turning a handle which simultaneously opens both suction and discharge valves in each pump. A vent near each injection valve is opened

to permit escape of air. The handle just mentioned may be turned far enough to open the suction valves only when it is desired to stop the engine.

LLOYD'S RULES

Engines built for marine use must comply with rigid governmental regulations. The best known of these are Lloyd's rules. These regulations specify materials used, broad types of construction and proportions for main engines and auxiliary engines and apparatus.

For instance the diameter of the crank-shaft is fixed by the following rule for multi-cylinder Diesel engines:

$$d = \sqrt[3]{D^2 + aL + bS}$$

where d = shaft diameter
D = cylinder bore
L = stroke
S = distance between inner edge of adjacent bearings.

a and b are constants furnished as part of the rules.

With the usual relation of stroke to diameter and width of crank arm, this formula gives a diameter of shaft about five-eighths of the cylinder bore.

TESTS

An extensive test of a two-cycle Diesel engine is reported in *Motor Ship* for October, 1922, pp. 772, 773. The test was reported by Professor W. H. Watkinson. The engine tested was of the Camellaird-Fullagar type. In each vertical power cylinder a pair of pistons move always in opposite directions and at equal velocities. The pistons over run ports at the end of the stroke. The port controlled by the upper piston admits scavenging air, while the burnt exhaust gas escapes through the port uncovered by the lower piston. This construction eliminates all valves except the small injection and starting valves.

The power cylinders are built in pairs, the lower power piston in each pair being connected by cross-rods to the upper piston in the other cylinder. This construction requires but one crank-pin for each power cylinder, though there are two power pistons in each cylinder.

The engine had two pairs of these power cylinders driving the same shaft. The bore was $18\frac{1}{2}$ inches, the stroke of each piston

25 inches, giving a total linear piston displacement in each cylinder of 50 inches. This construction evidently makes a very tall engine with heavy reciprocating masses, but also a well-balanced engine.

The ratio of scavenging piston displacement to power piston displacement was 1.715, and the ratio of the low-pressure compressor piston displacement to power piston displacement was 0.1245.

The oil was a heavy residual fuel oil of the following properties:

B.t.u. per pound (lower value)	18,175
Specific gravity	0.879
Flash point	183° F.
Viscosity at 63° F	58 seconds

Test No.	1	5	6	9*a*
M.e.p. all power pistons	89.0	59.5	66.6	92.8
Revolutions per minute	117.3	58.5	90.5	121.1
Indicated horse-power total	1420.0	473.0	818.0	1526.0
M.e.p. all scavenging pistons	2.86	0.72	.....	3.11
Indicated horse-power total, s.p.	78.2	.....	.....	87.6
M.e.p.—First-stage compressor	14.2	19.9	12.8	12.8
M.e.p.—Second-stage compressor	71.1	87.2	75.5	83.3
M.e.p.—Third-stage compressor	350.4	279.0	273.8	358.5
Total indicated horse-power compressor	90.9	51.3	60.7	97.2
Net indicated horse-power	1251.0	.....	.....	1341.0

The heat balance for test 9*a* is as follows:

Heat equivalent of delivered work	32.7	
Heat equivalent of engine losses	11.7	44.4
Heat carried away by jacket water	21.0	
Heat carried away by piston-cooling water	3.6	
Heat carried away in exhaust	24.8	
Heat lost by radiation	6.2	
Total	100.0	

Indicator cards for the power cylinders are given in Fig. 121. The values of n in the equation $PV^n = C$ were calculated from

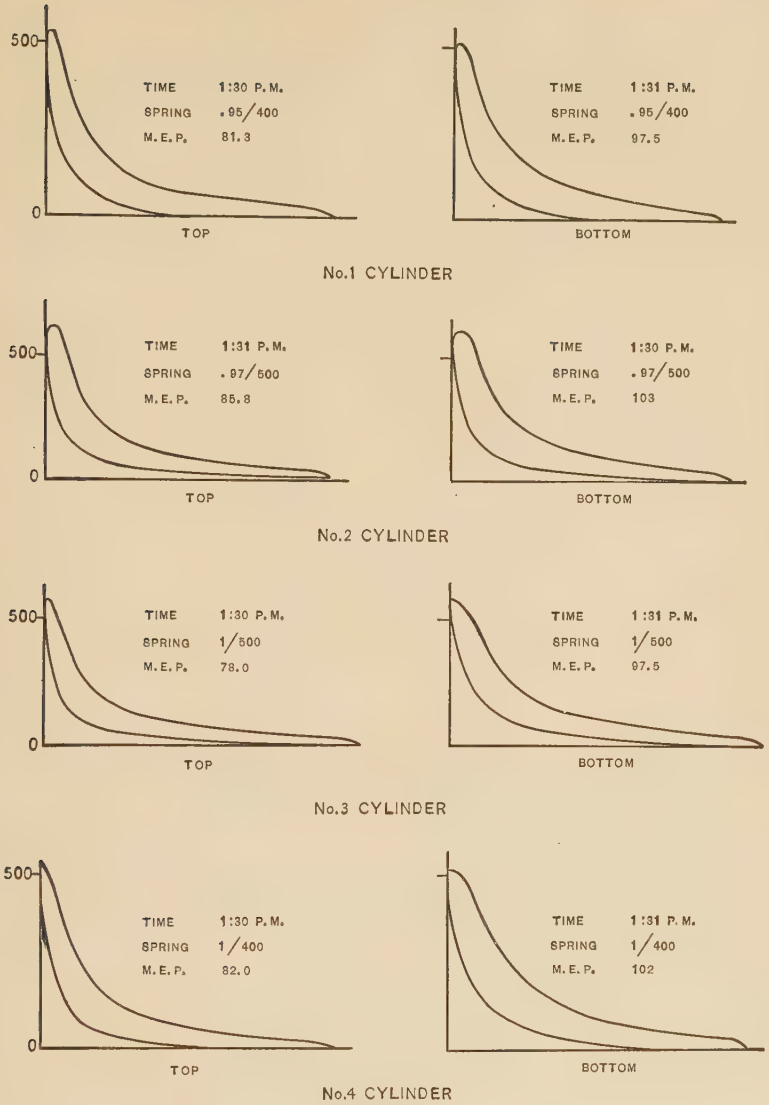


FIG. 121.—Indicator Cards from Marine Diesel Engine.

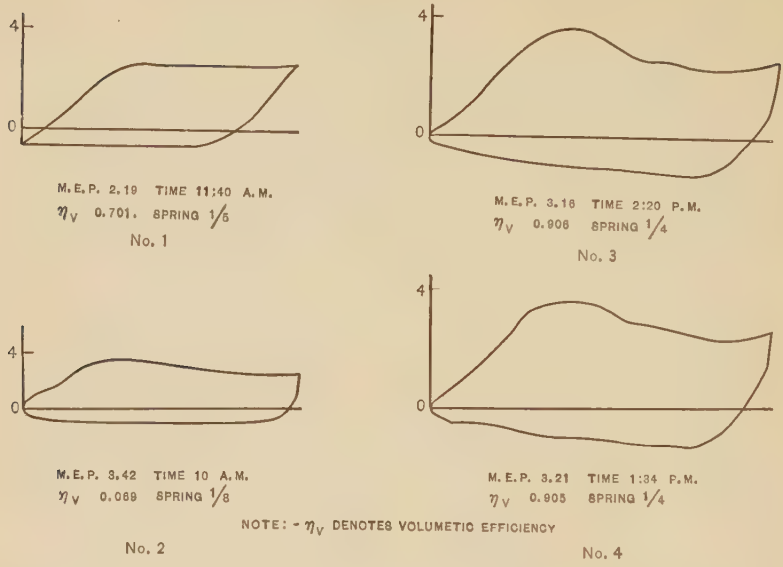


FIG. 122.—Scavenging Cylinder Cards, Marine Diesel Engine.

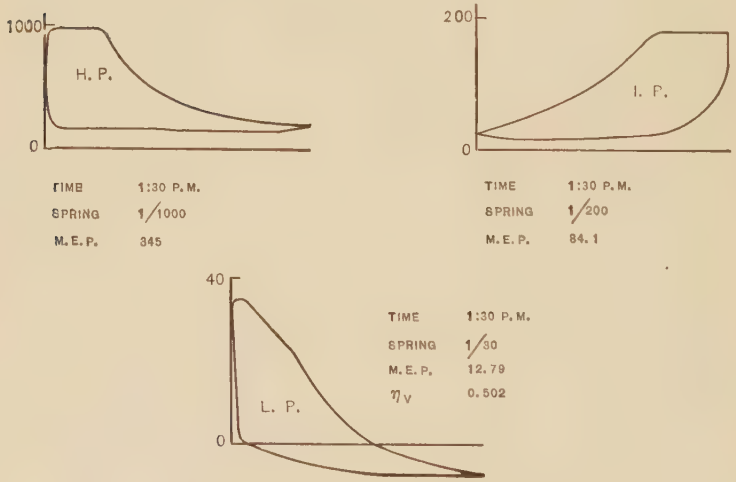


FIG. 123.—Air Compressor Cards, Marine Diesel Engine.

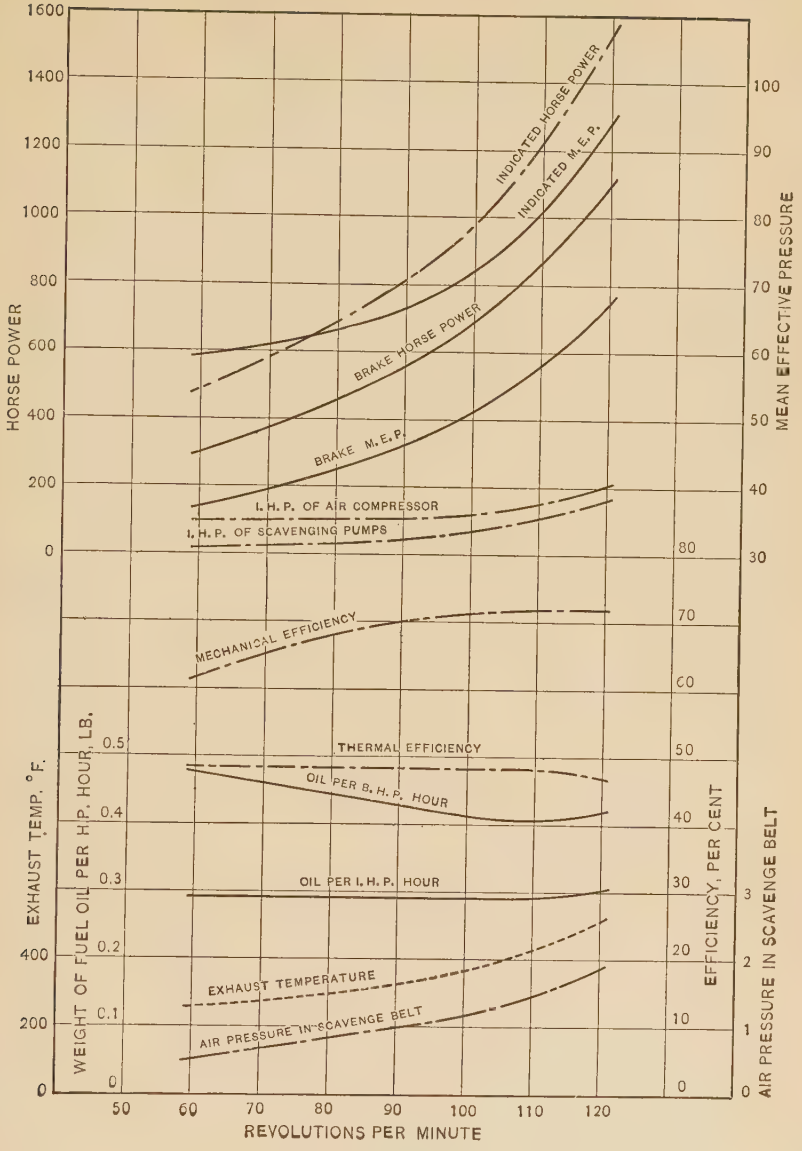

HORSE POWER
MEAN EFFECTIVE PRESSURE
INDICATED HORSE POWER
INDICATED M.E.P.
BRAKE HORSE POWER
BRAKE M.E.P.
I.H.P. OF AIR COMPRESSOR
I.H.P. OF SCAVENGING PUMPS
MECHANICAL EFFICIENCY
THERMAL EFFICIENCY
OIL PER B.H.P. HOUR
OIL PER I.H.P. HOUR
EXHAUST TEMPERATURE
AIR PRESSURE IN SCAVENGE BELT
EXHAUST TEMP. °F.
WEIGHT OF FUEL OIL PER H.P. HOUR, LB.
EFFICIENCY, PER CENT
AIR PRESSURE IN SCAVENGE BELT
REVOLUTIONS PER MINUTE

FIG. 124.

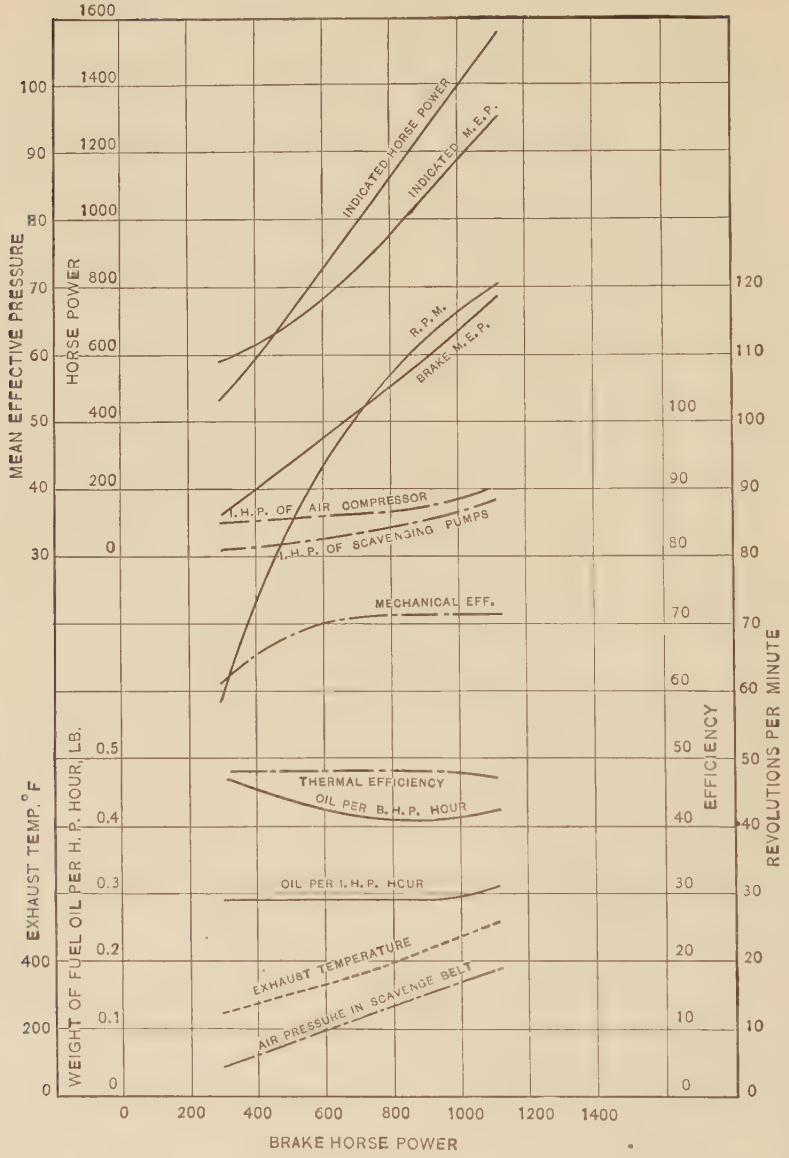

MEAN EFFECTIVE PRESSURE
HORSE POWER
EFFICIENCY
REVOLUTIONS PER MINUTE
EXHAUST TEMP. °F
WEIGHT OF FUEL OIL PER H.P. HOUR, LB.
BRAKE HORSE POWER
INDICATED HORSE POWER
INDICATED M.E.P.
R.P.M.
BRAKE M.E.P.
I.H.P. OF AIR COMPRESSOR
I.H.P. OF SCAVENGING PUMPS
MECHANICAL EFF.
THERMAL EFFICIENCY
OIL PER B.H.P. HOUR
OIL PER I.H.P. HOUR
EXHAUST TEMPERATURE
AIR PRESSURE IN SCAVENGE BELT

FIG. 125.

one set of cards for both compression and expansion line in the power cylinders. These values of n are given below:

Cylinder No.	n for Compression	n for Expansion
1	1.396	1.167
2	1.371	1.207
3	1.344	1.183
4	1.378	1.244

Indicator cards from the scavenging cylinders, Fig. 122, and from the air compressor follow, Fig. 123.

Charts taken from the results of various tests are shown in Figs. 124 and 125.

FIELD

The Diesel engine is, of all heat engines, the most economical in the use of fuel. The steam turbine and the reciprocating engine each use more than twice as much fuel per effective horse-power hour.

It is at present sold extensively only in moderate sizes. For land use the size limits are about 100 actual horse-power to 2000 actual horse-power. For marine use it is built from about 50 shaft horse-power to 6000 shaft horse-power.

Within these limits, where power is required for several thousand hours each year, with a load factor above 50 per cent, it is the best prime mover for the development of power.

As an auxiliary in a heating system its advantage is very much less. That is, in a heat and power plant where much more energy is needed in the form of heat than in the form of power, careful calculations would be required to decide on the best plant to install. For instance a factory where heat is required 7 months in the year might be better served by a Diesel engine or by a turbine and boilers, depending on the relative amounts of heat and power required in the whole year.

The Diesel power plant requires little if any more space than any other complete power plant. Turbines occupy only a fraction of the space of Diesel engines for the same power, but they require very many auxiliaries and a complete boiler plant in addition.

The fuel storage space for the Diesel power plant is very much smaller than for any other power plant.

The Diesel engine is a self-contained power unit and is far more quickly started than any of its competitors. Its stand-by fuel losses are very much less, or, rather, are absent.

The argument is sometimes advanced that the Diesel engine should not be purchased because the oil supply of the world, and especially of the United States, is rapidly failing.

It is true that the crest of production of oil from wells in the United States has almost been reached, but the total fuel oil resources of the world will certainly last for centuries. Every Diesel engine placed in service is likely to extend rather than limit the life of our fuel reserves.

On land the use of Diesel engines is increasing rapidly, especially in districts where fuel is expensive. Its limitation in size greatly restricts it here.

On the sea for moderate-sized freighters, up to 20,000 tons, and especially about 10,000 tons capacity, the Diesel engine is rapidly displacing both steam engines and steam turbines. Numerous steam-driven vessels are being converted to Diesel propulsion at present.

A fairly complete list of manufacturers of Diesel engines in America may be found in the latest edition of the Condensed Catalogues of Mechanical Equipment published annually by the American Society of Mechanical Engineers.

OIL ENGINES

The line dividing Diesel engines from other oil engines is so hard to define that different makers may call the same engine either Diesel or semi-Diesel. The use of 500 pounds per square inch compression with injection air at 900 to 1000 pounds per square inch always characterizes an engine as a Diesel. Other engines use the same high compression with solid injection. These are most often called Diesel engines.

Another type uses about 350 pounds per square inch compression with solid injection and combustion mainly at constant volume. These engines are properly called high-compression heavy oil engines but may also be called semi-Diesel engines.

The De La Vergne Machine Co. of New York City are the oldest builders of heavy oil engines in the United States. They

now manufacture true Diesel engines and several different types of high-compression oil engines.

Their type SI oil engine, otherwise known as pump injection oil engine is built in sizes of 100 and 150 horse-power per cylinder in the horizontal design and in 65 to 70 horse-power per cylinder in the vertical design.

Fig. 126 shows a vertical section through the cylinder of the horizontal type.

The engine is of the four-stroke cycle center crank type with split flywheel and counterweights bolted to crank-arms. The

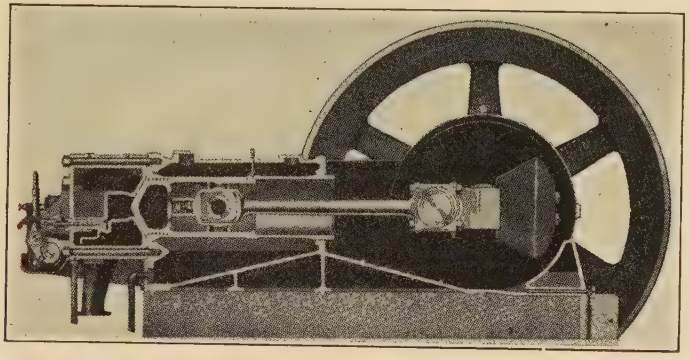


FIG. 126.—De La Vergne Type SI Oil Engine.

connecting-rod forging has a marine head at one end and is closed at the wrist-pin end. The piston is of the long trunk type with a conical head. The combustion chamber is thoroughly water cooled.

A cam-shaft runs across the end of the cylinder head and operates the inlet valve, the exhaust valve, and the air-starting valve. Two spray valves, not shown, enter the combustion chamber from opposite sides and the sprays, discharged through fine openings under a very high pressure, meet in the center.

Air pressure at 150 pounds per square inch is used for starting. Though the compression pressure is only 350 pounds per square inch no preliminary heating of any part is required. The temperature of compression is about 800° F. With air injection this temperature would be too low to cause spontaneous ignition. The location of the two jets or sprays is also an important advantage. The sprays, striking each other, do not readily deposit oil on the

walls of the combustion chamber, and with a slight lateral displacement a violent whirl should result.

No positively operated fuel valve is required. When the pump delivers oil it passes through an automatic valve near the spray nozzle. This always keeps the discharge pipe under the high

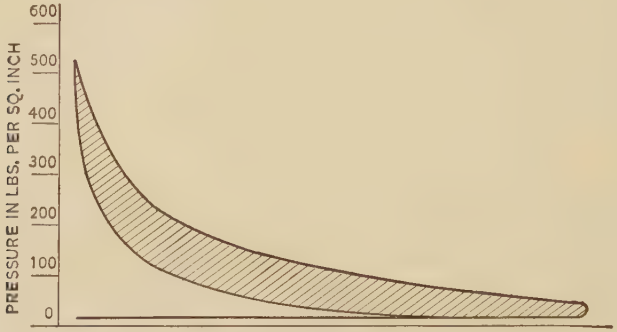


Fig. 127.—Indicator Card. De La Vergne Type SI Oil Engine.

pressure. The governor controls the speed by opening a by-pass in the discharge line, permitting some oil to return to the suction side of the pump.

Fig. 128.—Maker's Guarantee and Expected Performance Curves. De La Vergne Type SI Oil Engine.

The absence of an injection air compressor raises the mechanical efficiency 5 to 10 per cent above that of a Diesel engine.

Fig. 127 shows a typical indicator card from this engine.

Fig. 128 gives the maker's guarantee and expected performance curves. It will be noted that the guarantee is stated

to be 0.45 pound per actual horse-power at full load and 0.50 pound at half load.

The fuels for which the engine is designed are: Crude oil, distillate or fuel oil produced in the United States or Mexico.

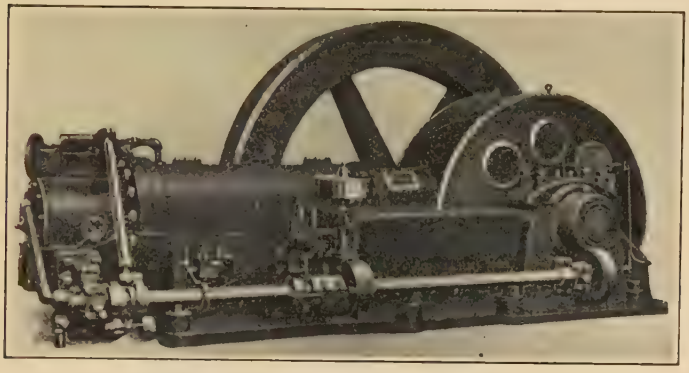


Fig. 129.—150 H.P. De La Vergne Type SI Oil Engine.

Fig. 129 shows an outside view of the 150 horse-power size of this type. It will be noted that the inlet and exhaust valves are top and bottom. The very massive construction is apparent.

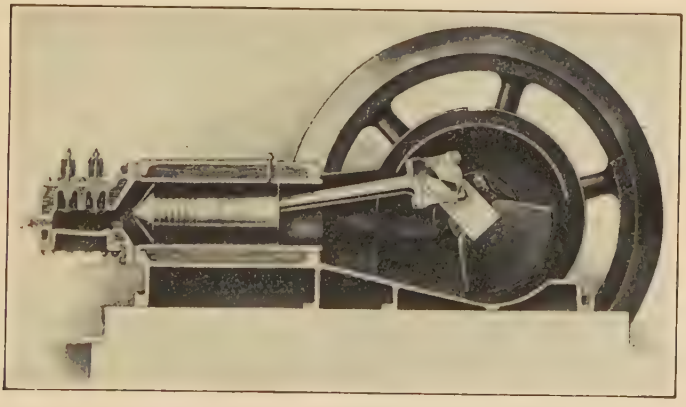


Fig. 130.—De La Vergne Type DH Oil Engine.

Note the governor controlling the fuel pump which is alongside. One of the spray valves is seen at center of the head. A force feed lubricating pump is about half way between the fuel pump and

air cam. The diagonal pipe running across the end of the cylinder head is the air pipe for starting. A valve on the end of the cam-shaft times the air admission in starting.

The engine is built in sizes from 100 to 300 b.h.p.

Type DH.—Fig. 130 shows a sectional side elevation of this type which is also built by the De La Vergne Machine Company. It is also a four-stroke cycle engine. It is of lighter construction than the SI, built in sizes of 48 and 65 b.h.p. per cylinder. The largest engine is a twin of 130 b.h.p.

The compression pressure is a little less than 300 pounds and the maximum pressure 450 pounds.

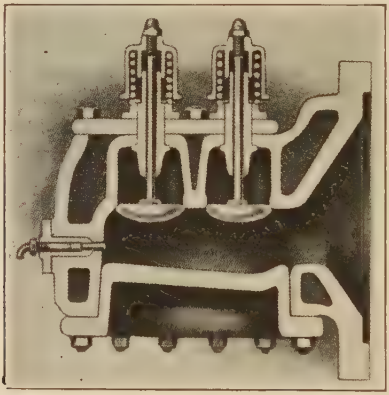


Fig. 131.—Section through Vaporizer De La Vergne Type DH Oil Engine.

Fig. 131 shows an enlarged section through the cylinder head. It will be noted that a "hot spot" or uncooled part of the cylinder head is used to assist in vaporizing the oil. Solid injection is used with a single nozzle. It is necessary to heat the vaporizer before starting this engine.

The engine will operate on any commercial crude oil, distillate or fuel oil. The guarantee is: full load 0.5 pound per b.h.p., half load 0.52 pound per b.h.p.

Price-Rathbun Oil Engines.—The Ingersoll-Rand Co. build stationary four-cycle oil engines of three, four, or six cylinders in sizes from 100 b.h.p. to 1000 b.h.p. Marine oil engines are built up to 750 b.h.p.

Pump injection, or solid injection, is used with a compression pressure of about 350 pounds per square inch. Fig. 132 is an

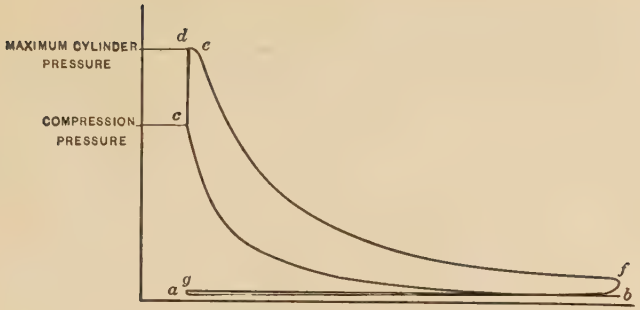


FIG. 132.—Indicator Diagram Price-Rathbun Oil Engine.

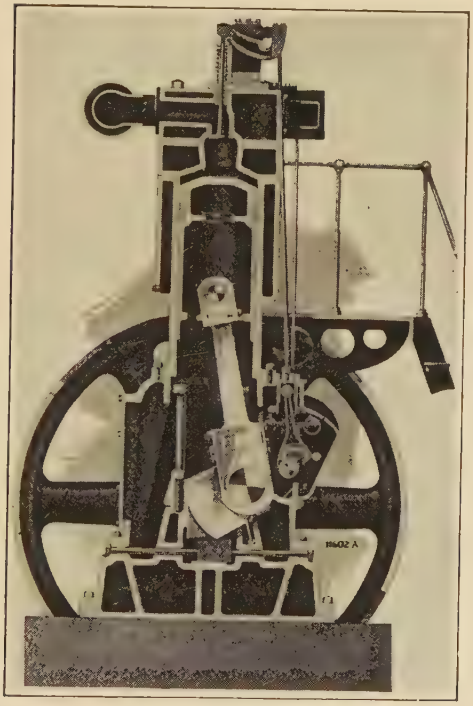


FIG. 133.—Vertical Section Price-Rathbun Oil Engine.

indicator card from this engine. The engines are four-stroke cycle, and the card has the rounded "toe" characteristic of this type. The maximum pressure is about 500 pounds per square inch.

Fig. 133 shows a sectional elevation of the engine and brings out the shape of the combustion chamber. The double head of the piston, with the upper head conical is always used. It will

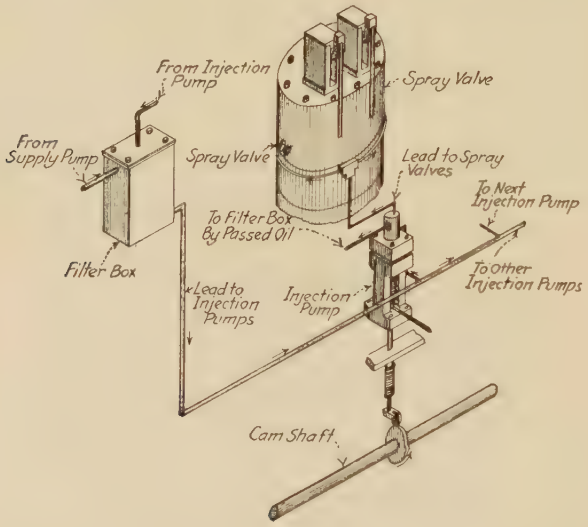


Fig. 134.—Fuel-oil Distributing System, Price-Rathbun Oil Engine.

be noted that the "valve" operating shaft is located nearly on a level with the main bearing and that it operates each valve through an eccentric on the shaft and a rocking cam lever above the valve. This gives quiet operation and the great leverage provided at valve opening greatly reduces the strain on the long push rod.

Fig. 134 shows the fuel oil distributing system. A fuel pump operated by a cam is provided for each cylinder. During the injection period the plunger is lifted discharging the oil into the cylinder through two nozzles or spray valves. The finely divided oil jets impinge on each other, instead of against the walls of the combustion chamber. Ignition takes place as soon as oil starts to enter. The sudden rush of air from the space between piston and head gives this double jet a violent turbulent motion and

effectively distributes the burning oil mist throughout the combustion chamber and effects complete combustion.

A lever above the pump and moved by the pump plunger operates a by-pass valve regulating the flow of oil into the nozzle. A

Fig. 135.—Fuel-oil Injection Pump, Price-Rathbun Oil Engine.

fulcrum under control of the governor changes the travel and timing of the lever and thus regulates the amount of oil sprayed into the cylinder.

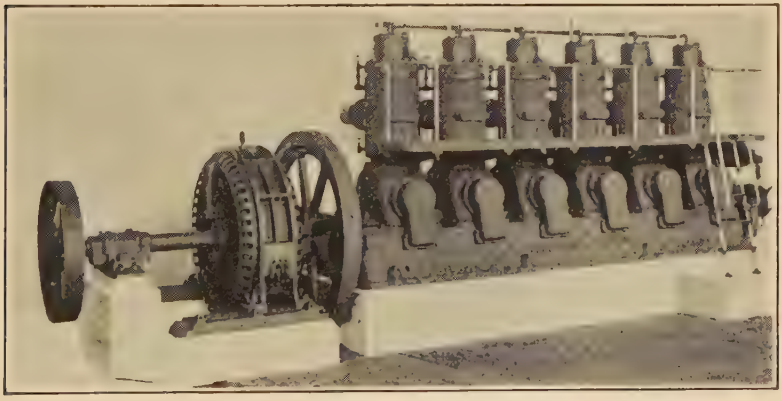


Fig. 136.—Fairbanks-Morse 300 H.P. Oil Engine.

Fig. 135 shows a photograph of the pump. The handle for operating the pump is in front. The by-pass lever and fulcrum are shown at the top.

The fuel consumption in pounds of oil per brake horse-power for a 300 horse-power engine given for various loads is taken from the figures of the manufacturer.

Load, Per Cent	100	90	80	70	60	50	40	30	20
Pounds oil per Brake horse-power per hour.....	0.40	0.39	0.39	0.41	0.43	0.47	0.52	0.62	0.83

The Fairbanks-Morse Co. build a two-stroke cycle moderate pressure oil engine. This is called their "Y" engine. Solid injection is used. The cylinder head is water jacketed. As the engine must be warm to operate a small tube heated by a blow-torch is provided for starting. This operation takes only a few minutes.

Fig. 136 shows a six-cylinder 300 b.h.p. engine direct connected to a dynamo. The oil pipes and spray nozzles will be seen at the top of each cylinder. The fuel oil pump is located at the end of the crank-shaft furthest from the dynamo. The fuel guarantee is 0.6 pound oil per b.h.p. hour at full load and 0.7 pound oil per b.h.p. hour at half load.

CHAPTER XVI

THE COST OF POWER GENERATED BY INTERNAL COMBUSTION ENGINES

Plants of Various Sizes—Gas, Gasoline and Oil Engine Plants

General.—It is difficult to analyze the cost of power generated in a small plant, and arrive at a conclusion that is very near the truth, unless every condition is known beforehand. Since conditions vary with every plant, it is obviously impossible to make a detailed estimate, or an accurate estimate that will cover all small plants or all large plants. Certain assumptions may be made and on these assumptions certain estimates may be made, but the results thus obtained should be used with caution.

In order to show how estimates may be made, the author will discuss, in this chapter, certain costs that occur in small plants of different capacities. The discussion that follows will treat of isolated plants and the cost of power generated for a specific purpose, not for general distribution. Three types of engines will be taken up, steam, gas and oil. Steam turbines will not be touched upon as that would involve problems concerned only with the relative merits of the turbine and the steam engine.

The form of power to be generated is of importance. In order to have uniformity, we will assume that in each case the engine is direct connected to a direct-current generator, the current to be used for lighting and power in a manufacturing plant, during a working day of 10 hours, 6 days per week, or 300 days per year. The sizes of the plants selected will be 20, 100, 250 and 500 kilowatts. In no case will the cost of the land or building be included, as it is impossible to get even an approximate figure that would be of value for this item.

In general, the costs of the machines were secured from manufacturers. The prices in each case represent a mean of those submitted, in some cases, five or six, in others only two. The cost of the foundation in each case was based on the floor space, and of the piping, on the cost of piping in similar plants. The

cost of the machines in each case includes erecting, freight for three or four hundred miles, cartage, etc.

The "fixed charges" were apportioned for the steam and producer gas plants as follows:

	Per Cent
Interest on investment	6
Depreciation	6
Repairs	2
Taxes	1
Insurance	1
Total	16

The depreciation will not be the same on all parts of the plant, nor will the cost of repairs be uniform over the whole plant, but such figures as were selected represent the average depreciation and repairs on the whole plant. For the high-pressure oil-engines, the depreciation will be assumed to be $6\frac{1}{2}$ per cent and the repairs $2\frac{1}{2}$ per cent, for it has been the experience that these plants cost more to keep in repair than plants where working pressures are not so high.

Since no plant can be expected to run at full load all the time, we will assume a load factor of 75 per cent. This factor would be high for a plant selling power, in fact, it would be a maximum for the type of plant under consideration.

The wages of a first-class engineer for the larger plants was assumed to be $6 per day, firemen $4.50 per day, and for second-class engineers, for the smaller plants, $4.50 per day. In the smaller plants and in all oil-engine plants, there was no allowance made for a night man.

The coal was based on 13,000 and oil on 19,000 B.t.u. per pound. The price of water was assumed to be 10 cents per 1000 gallons, lubricating oil 22 cents per gallon and cylinder oil 30 cents per gallon.

Twenty-kilowatt Plant.—The approximate costs of three different types of 20-kilowatt plants are given below. For the steam plant the cost is based on a high-speed, horizontal engine, running non-condensing. The type of boiler selected was the vertical fire-tube on account of the low cost, being $125 less than the horizontal return-tubular type.

The coal consumption was based on, first, efficiency of boiler, 60 per cent; and second, steam consumption of engine of 35

pounds per indicated horse-power per hour. An additional 15 per cent was allowed on the coal for banking and losses.

The water cost was based on the above, with an allowance of 15 per cent for leakage, etc.

For the gas engine plant, an anthracite producer is selected as being the best adapted for the work. The coal cost is based on 2 pounds per horse-power hour including stand-by losses. The care of a plant of this size will not require the full time of one man; in fact less than half of it; hence the labor cost of $540. The cost of water was based on 2 cubic feet per kilowatt hour for the entire plant, fresh water to be used continuously.

For the oil engine plant a low-pressure oil engine is selected as being better suited for this small plant than a high-pressure engine. The fuel cost is based on 1 pound of oil per brake horse-power hour, and the cost of water on a consumption of 5½ gallons per horse-power hour.

A table showing the itemized cost of each of the three particular types of plant follows:

Cost of Power—20-kilowatt Plant

	Steam plant	Producer-Gas plant	Low-pressure oil-engine plant
Brake horse-power	32	32	32
Indicated horse-power	35	.....	.....
Cost of engine and foundation	$1,200	$1,800[1]	$2,400[2]
Generator and switchboard	900	900	900
Boiler, foundation and piping	600	.....	.....
Producer and washer	.....	1,000	.....
Total	$2,700	$3,700	$3,300
Yearly cost:			
Coal at $6 per ton	1,320	450	.....
Oil (fuel) at 6 cents per gallon	.....	.....	600
Labor	1,350	540	360
Fixed charges	450	595	530
Oil, waste and supplies	150	160	150
Water	50	110	40
Total	$3,320	$1,855	$1,680
Cost per kilowatt hour, cents	7.40	4.13	3.74

[1] Air compressor included in this item.

[2] Air compressor and oil tank included in this item.

In order to show the varying cost of power with different fuel values, power-cost curves have been plotted with cost of fuel as abscissæ and cost in cents per kilowatt hour as ordinates. The curves for the 20-kilowatt plant are shown in Fig. 137. The units on the horizontal scale are "dollars per ton" for the coal cost of the steam- and gas-engines, and "cents per gallon" for the oil-engine. The fuel cost for the steam-engine is a greater percentage of the total cost than for the gas-engine, hence the rapid rise in the line representing the steam-engine cost.

One Hundred-kilowatt Plant.—For the steam plant a high-speed, non-condensing engine is selected, as the first cost and fixed charges of a condensing engine of this size would be prohibitive. The boiler selected is of standard water-tube design. The coal and water cost are based on boiler efficiency of 60 per cent, engine to use 30 pounds of steam per indicated horse-power per hour, with an additional allowance of 15 per cent for losses.

For the producer plant a suction producer operating on anthracite coal is assumed as the most convenient and the cheapest. A bituminous coal producer of this size would cost $4500 to install compared with $3300 for the anthracite producer. The engine cost is based on a three-cylinder, vertical unit. The coal cost is based on $1\frac{1}{2}$ pounds per brake horse-power hour at 75 per cent load factor, 10 per cent being added for stand-by losses. The water rate is assumed to be 15 gallons per horse-power hour, 5 for the engine and 10 for the producer and scrubber, using the water only once.

The 100-kilowatt oil engine selected for use in this discussion is of the high-pressure type, either Diesel or semi-Diesel. The item of engine cost includes all auxiliaries, tanks, oil pump, air compressor, etc. The fuel cost is based on 0.55 pound per brake horse-power hour.

The comparison of the 100-kilowatt steam, gas and oil engines appears in the following table:

If a low-pressure oil engine had been used instead of a high-pressure, the cost of the plant would have been $5000 less. In that case the yearly fixed charges would have been 16 per cent on $14,550, about $2328. The saving on this item would thus be $776 per year. The oil consumption for the low-pressure engine would be about 46,000 gallons, costing $2760 against $1500 or the high-pressure engine, the difference being $1260 in favor of

the high-pressure engine. The net difference is $484 in favor of the high-pressure engine. But if oil costs 3 cents per gallon, the balance would have been $396, and in favor of the low-pressure engine. This interesting point is brought out to show that as

COST OF POWER—100 KILOWATT PLANT

	Steam plant	Producer gas plant	High-pressure oil-engine plant
Brake horse-power	150	150	150
Cost of engine and foundation	$ 4,500	$ 8,500	$17,000
Boiler, pump and piping erected	6,500	.....	.....
Producer and piping	.....	3,300	.....
Generator and switchboard	2,400	2,400	2,400
Steel stack and foundation	750	.....	.....
Total	$14,150	$14,200	$19,400
Cost per year:			
Coal at $6 a ton	$ 4,800	$1,650	.....
Fuel oil at 6 cents per gallon	.....	.....	$1,500
Attendance	2,700	1,350	1,350
Fixed charges	2,264	2,272	3,104
Oil, waste and supplies	200	200	200
Water	150	500	170
Total	$10,114	$5,972	$6,324
Cost per kilowatt hour, cents	4.50	2.65	2.81

the price of fuel increases, it is a paying investment to put money into more efficient equipment.

In order to show the effect of increased price of fuel on power cost, Fig. 138 is shown here—a chart similar to the one in Fig. 137, except that it is for the 100-kilowatt plant.

Two Hundred fifty-kilowatt Plant.—For this size plant the steam-engine is assumed to be a non-condensing, tandem-compound Corliss. The steam consumption of this engine would be about 22 pounds per horse-power hour. This figure was used to get the size of the boilers and the coal consumption. In working out the coal consumption, 10 per cent was allowed for auxiliaries and 10 per cent for stand-by losses.

The gas engine used for the 250-kilowatt plant is of the four-cylinder vertical type, gas to be furnished by one anthracite pro-

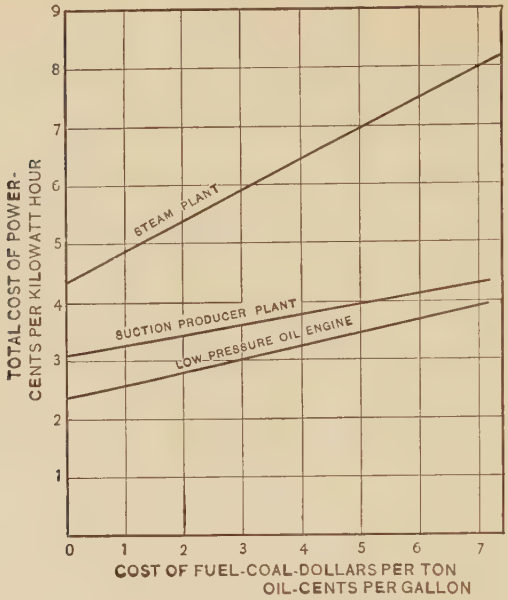


FIG. 137.—Cost of Power Generated by 20-kilowatt Plants.

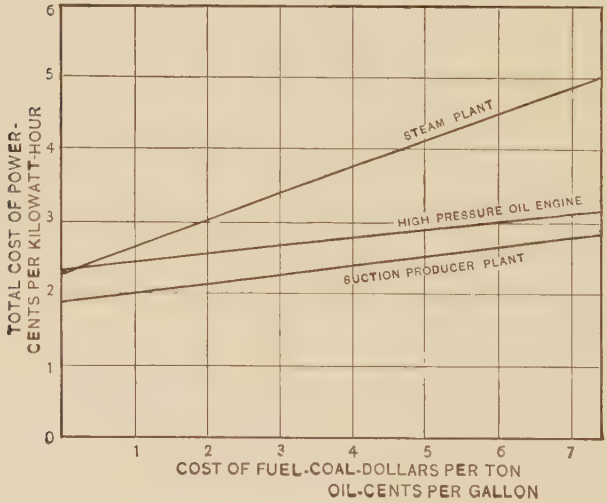


FIG. 138.—Cost of Power Generated by 100-kilowatt Plants.

ducer. The coal cost is based on 1.5 pounds of coal per brake horse-power hour, including stand-by losses. The water charge is based on $7\frac{1}{2}$ gallons per brake horse-power hour, using the scrubbing water over and over.

The oil engine selected is of the high-pressure type, with a consumption of one-half pound of oil per brake horse-power hour.

It is interesting to note that the 250-kilowatt steam plant is the cheapest to operate with fuels at low cost. In Fig. 139 it will be noticed that the steam engine line crosses the suction producer line when coal is worth $1.20 per ton, and the steam and oil engine lines cross with coal at $1.90 per ton and oil at 1.90 cents per gallon. It is noticeable, also, that the line for the steam plant is less steep than for the 20- and 100-kilowatt plants. This is because the fuel cost is less in proportion to the total cost than in the smaller plants.

Cost of Power—250-kilowatt Plant

	Steam plant	Producer gas plant	High-pressure oil-engine plant
Brake horse-power	365	365	365
Cost of engines, foundations, etc.	$ 8,775	$19,500	$40,000
Boiler or producer and piping	8,400	7,685	.....
Steel stack and flues	900	.....	.....
Generator, switchboard and wiring	6,000	6,000	6,000
Total	$24,075	$33,185	$46,000
Cost per year:			
Coal at $6 a ton	$ 9,000	$ 3,750	.....
Oil at 6 cents a gallon	.....	.....	$ 3,420
Attendance	3,150	2,700	1,600
Fixed charges	3,852	5,310	7,360
Oil, waste and supplies	300	300	300
Water	335	600	400
Total	$16,637	$12,660	$13,080
Cost per kilowatt hour, cents	2.95	2.25	2.32

Five Hundred-kilowatt Plant.—For the steam plant of this capacity a compound-condensing Corliss engine is assumed. The steam consumption would be about 18 pounds per indicated horse-power hour for the main engine, with 20 per cent additional

for auxiliaries and stand-by losses. It is also assumed that condensing water can be had at no expense except for pumping, which is allowed for in the cost of the plant, and steam for auxiliaries.

For the producer plant a bituminous pressure producer is assumed for several reasons, one of which is that the size is large for

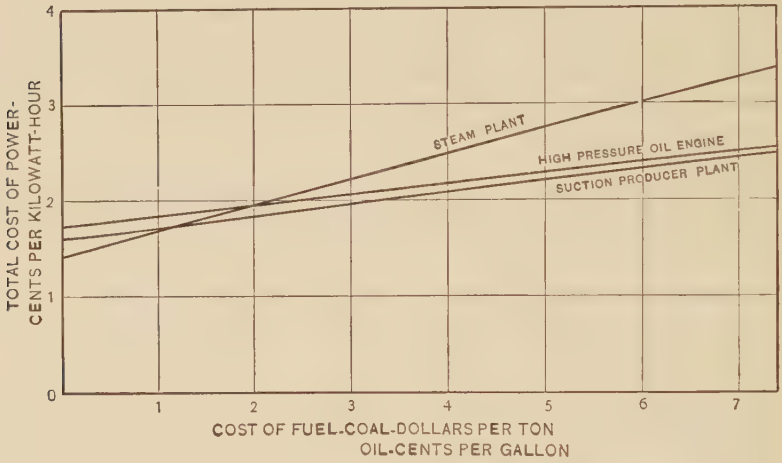


Fig. 139.—Cost of Power Generated by 250-kilowatt Plants.

a suction producer, and in this discussion we are limited to one unit in each case for the sake of uniformity. The bituminous producer is well adapted to this size of unit and it is well to point out the possibilities of such an installation. The cost is more than for a plain suction producer, but the efficiency is probably higher; certainly the pressure type is more flexible.

The coal cost is based on 1.25 pounds of coal per horse-power hour, including all stand-by losses. The water is based on $7\frac{1}{2}$ gallons per horse-power hour for the engine, producer and scrubber. In this case the water for the scrubber will have to be used over and over. If this is not done the water consumption will be doubled. In order to make it possible to use this water over and over, it should be cooled by spray nozzles after being run into a settling tank where the dust from the scrubber is deposited. The cost of the tank, nozzles, etc., has been taken care of in the cost of producers and auxiliaries.

For the oil engine the selection is a Diesel type, the cost including air compressor, oil tanks and pumps, piping, etc.

COST OF POWER—500-KILOWATT PLANT

	Steam plant	Pressure producer gas plant	High-pressure oil-engine plant
Brake horse-power	725	725	725
Cost of engine, foundation, etc.	$14,000	$37,500	$75,000
Boilers and pumps	15,000	.....	.....
Stack and flues	1,800	.....	.....
Condenser	3,000	.....	.....
Producer and auxiliaries	.....	19,500	.....
Generator, switchboard, etc.	14,000	14,000	14,000
Total	$47,800	$71,000	$89,750
Cost per year:			
Coal at $6 a ton	$15,000	$ 6,120	.....
Fuel oil at 6 cents a gallon	.....	.....	$ 6,720
Attendance	4,500	4,500	1,800
Fixed charges	7,648	11,360	14,360
Oil, waste and supplies	500	500	500
Water	420	1,000	840
Total	$28,068	$23,480	$24,220
Cost per kilowatt hour, cents	2.50	2.09	2.15

The diagrams showing cost of power in the different 500-kilowatt plants with varying cost of fuel are given in Fig. 140.

Other Fuels.—While the foregoing tables and diagrams cover the cost of power generated by engines using the more common fuels, in some cases other fuels may be used for special reasons. Among these fuels are natural gas, illuminating gas, gasoline and kerosene.

In order to give some idea of the cost of power generated by these fuels, a specific plant, 100 kilowatts capacity, will be discussed here.

The cost of the two gaseous fuels is based on an efficiency of 25 per cent. For the natural gas plant this means a consumption of 11.3 cubic feet per horse-power hour with gas having a heating value of 900 B.t.u. per cubic foot. For illuminating gas the figures are 17 cubic feet with gas of 600 B.t.u. per cubic foot.

The cost of gasoline is based on an efficiency of 20 per cent, assuming the gasoline to be used in a carburetor. The fuel con-

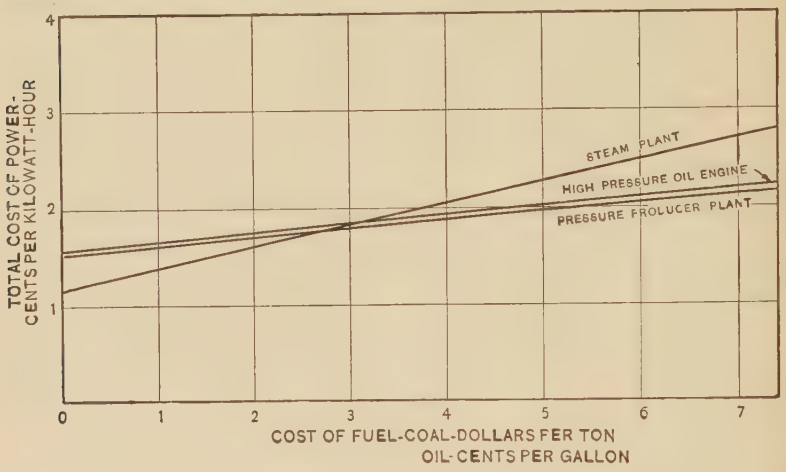


FIG. 140.—Cost of Power Generated by 500-kilowatt Plants.

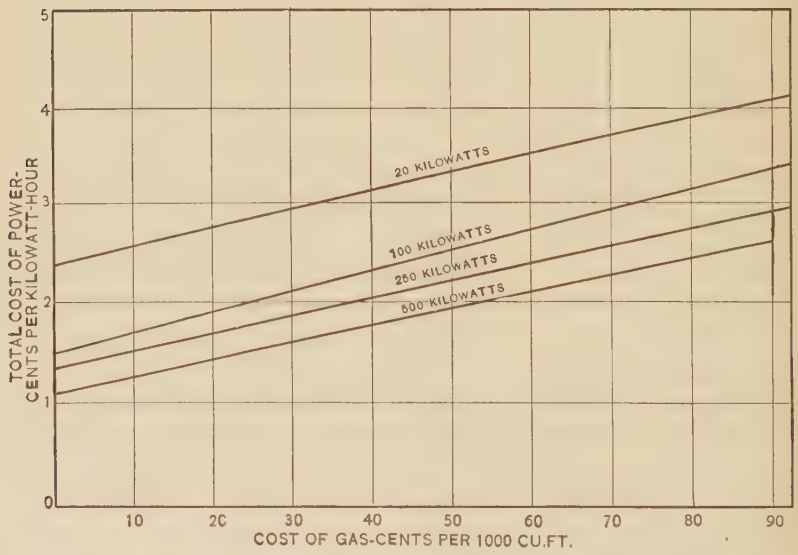


FIG. 141.—Cost of Power Generated by Plants Using Natural Gas as Fuel.

sumption with carbureted kerosene would be practically the same.

The natural gas cost is based on 40 cents per thousand cubic feet, illuminating at $1, and gasoline at 20 cents per gallon.

Figs. 141 and 142 illustrate the cost of power generated by natural gas and illuminating gas respectively, in various size units,

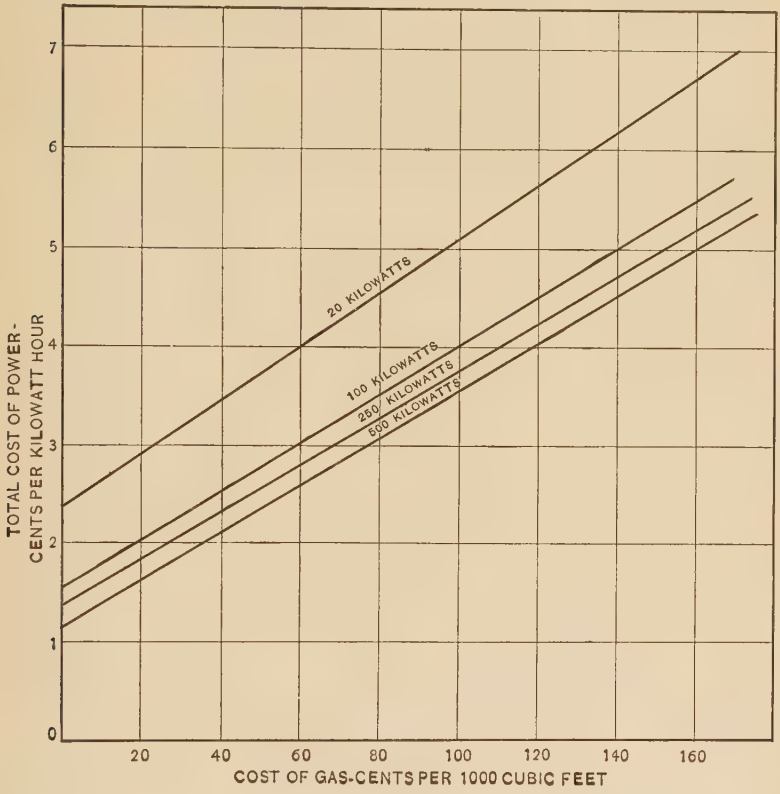


FIG. 142.—Cost of Power Generated by Plants Using Illuminating Gas as Fuel.

these costs being exclusive of land and buildings. In many cases the cost of land and buildings does not amount to much for plants of this size, as the floor space required for vertical engines is small, and auxiliaries do not take up much room.

Up to 40 or 50 cents per thousand cubic feet, natural gas makes an ideal fuel for the gas engine as it is clean, high in heating value, and the supply is usually reliable. In some districts the pressure

falls appreciably in severe weather in the winter, particularly in the early evening hours. In such localities provision must be made for other power at that time.

Cost of Power—100-kilowatt Plant

	Natural gas	Illuminating gas	Carbureted gasoline
Brake horse-power	150	150	150
Cost of engine, foundation and piping	$ 9,000	$ 9,000	$ 9,500
Generator and switchboard	2,400	2,400	2,400
Total	$11,400	$11,400	$11,900
Cost per year:			
Gas or gasoline	$ 1,534	$ 5,740	$ 7,960
Attendance	1,350	1,350	1,350
Fixed charges	1,824	1,824	1,904
Oil, waste and supplies	200	200	200
Water	170	170	170
Total	$ 5,078	$ 9,284	$11,584
Cost per kilowatt-hour, cents	2.26	4.13	5.15

Duplicate Units.—In each case of the foregoing discussion one unit was assumed to constitute the entire plant. There are many places where such an arrangement would be inadvisable; in fact in the majority of cases it would be better to have two units instead of one. The reason for this is not particularly the unreliability of one unit, but if the load varies greatly at different periods of the day or at different seasons, it would pay to put in two or more units so that each might carry a large percentage of its rated capacity whenever it is in operation. The advantages and disadvantages of such an installation would have to be investigated for each individual plant and settled in a way best suited to meet local conditions. In general, it might be said that the low efficiency of gas engines at light loads necessitates running each unit at a load near its rating in order to obtain the best results in fuel consumption. In plants where the power generated is to be used largely for lighting, or to operate non-reversing motors, it might be advisable to install alternating-current generators. In general this will lessen the cost of the generators, but add to the cost of switches and exciter equipment.

To have Next time.

CHAPTER XVII

POWER, EFFICIENCY, SPEED AND SIZE OF ENGINES

Brake and Indicated Horse-power

Power.—The engineering definition of power is the rate at which work can be done. Thus, an engine that can do 33,000 foot-pounds of work in one minute is said to be one horse-power. Obviously, the same engine can do 550 foot-pounds of work per second.

Brake Horse-power. — If the power of any engine is expressed in terms of the horse-power actually delivered, that is, power that may be used to do actual work, it is called the **brake horse-power,** or the power that may be absorbed by a prony brake.

Indicated Horse-power. — The power that is actually generated in the cylinder of an engine is greater than the power delivered by the engine to do useful work. The ratio of these two powers is called the mechanical efficiency of the engine. Thus,

$$\frac{\text{Brake horse-power}}{\text{Indicated horse-power}} = \text{mechanical efficiency.}$$

The difference is the power absorbed in friction, in opening valves, operating governing mechanism, oil and fuel pumps, igniters, etc. The frictional horse-power is often taken as the indicated horse-power at no load. This is not quite true as the friction work of bearings, crosshead slides, pumps, etc., will be greater at full loads than at lighter loads.

The **mechanical efficiency** of an engine is zero at no load and increases to a maximum at the maximum horse-power. On account of the great number of auxiliaries, the mechanical efficiency of a gas engine is lower than that of a steam engine. While the efficiency of a steam engine may be as high as 95 per cent, that of a gas engine seldom runs above 90 per cent at full load.

It is clear that the more power that is required to take care of the auxiliaries of a gas engine, the less will be the mechanical efficiency. In a Diesel engine the power required for the air compressor which furnishes the high-pressure air for the fuel injection must be charged to the engine. This decreases the useful

work for a given indicated horse-power and cuts down the efficiency in proportion.

Likewise, the power required to do the scavenging on a two-cycle engine cuts down the useful output and hence the mechanical efficiency.

Measuring the Output. — The output of a gas engine, that is the brake horse-power, may be measured in many ways. For small engines, up to 200 horse-power, the best way is to use a prony brake for comparative or approximate results. The prony brake is not particularly accurate, neither is it suitable for large powers. In some cases it is impossible to get a suitable brake pulley on the engine shaft, in which case some other means must be used to measure the output of the engine.

When an engine is built to be direct connected to an electric generator, the best method of measuring the engine output is to measure the electrical output and correct it for the efficiency of the generator, thus obtaining the brake horse-power of the engine. The efficiency of an electrical generator may be secured more accurately than the readings and measurements of a prony brake, so that in general the latter method is much to be preferred rather than the former.

Where engines are tested by manufacturers, carefully calibrated generators are coupled up to the engines and the efficiencies found for different loads. Or, in many cases, the engines are connected to fans or blowers instead of electric generators. Either method gives a means of varying the output over a wide range and keeping the load steady at each point, a difficult thing to do with the prony brake.

The Prony Brake. — Prony brakes are made in a variety of forms, one of which is shown in Fig. 143. The brake wheel W should be made with flanges on the inside of the rim to retain water. The band B is best when made of a woven belting. This band is riveted to a plate at either end, the plates being forged to the adjusting screws S. The arm A has attached to it a block E cut out to the radius of the wheel. The adjusting nut D is most convenient when made in the form of a hand wheel.

Power is absorbed by friction of the band on the wheel, and in order to prevent burning, this band should be continually saturated with lubricating oil. The heat generated is carried away by the vaporization of the water in the wheel.

With a brake of this type, having a band three inches wide, the author has had no trouble in absorbing 15 horse-power where the peripheral speed was as high as 4800 feet per minute.

The horse-power of a prony brake is found by the equation

$$\text{Brake horse-power} = \frac{2\pi FRN}{33{,}000}.$$

Since the denominator is in foot-pounds per minute, it is obvious that the units in the numerator must be selected to give foot-pounds per minute in the numerator. Since the distance must be in feet, R must be in feet, giving as the distance $2\pi R$ for one revolution, or $2\pi RN$ as the distance moved through per minute. If the force F, measured on the scales, is in pounds, then $2\pi FRN$ will be in foot-pounds per minute as required.

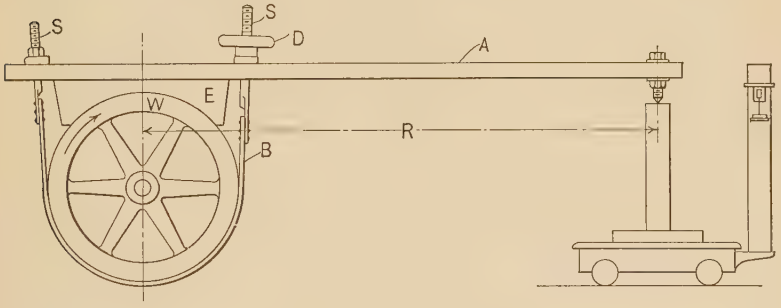


FIG. 143.— Prony Brake.

It is sometimes difficult for any one not familiar with the prony brake to thoroughly understand that, even though the force at the scales does not move we get the same effect as if it did, since the movement occurs at the periphery of the wheel. The force if applied at the rim of the wheel would have to be greater than the force at the radius R by the ratio $\frac{R}{\text{rad. wheel}}$. This would give the same work per minute as the distance would be decreased by the ratio $\frac{\text{rad. wheel}}{R}$.

Another way to look at it is to assume that the band is tight on the brake wheel and that the force F moves in a circle of radius R. Then, clearly, the distance moved through is $2\pi RN$ feet per minute.

Another point which is necessary to keep in mind is that this same formula is applicable to a prony brake wherever used, and that the type of gas engine makes no change in the formula. This, we will see later, is not true of the formula for the indicated horse-power.

The factor F in the brake formula is the net load at R. The zero, or non-effective load, may be found by turning the engine slowly forward with the band loosed, and taking the weight; then in the opposite direction, taking the weight. The mean of these two weights is the zero load, and should be subtracted from the scale load in order to get the value F.

The Alden brake is one in which the rubbing surfaces are discs separated by a film of oil, and the heat is absorbed by water under pressure, which produces the friction. It is constructed by fastening a disc of cast iron to the power shaft; this disc revolves between two sheets of thin copper joined at their outer edges, from which it is separated by a bath of oil. Outside the copper sheets on either side is a chamber which is connected to the water supply. The water flows through this chamber constantly, maintaining a uniform temperature. Any pressure in the chamber causes the discs to press against the revolving plate, producing friction which tends to turn the copper discs. As these are rigidly connected to the outside cast-iron casing and brake arm, the turning effect can be balanced and measured the same as in an ordinary prony brake. The pressure of the water is automatically regulated by a valve, which is partially closed if the brake arm rises above the horizontal and is partially opened if the arm falls below the horizontal. This brake with a constant head of water gives very close regulation. Another advantage is that it can be used for large powers.

The Sprague Dynamometer.—The Sprague electric absorption dynamometer is an apparatus capable of the highest degree of accuracy. It comprises an armature field magnet combination which may be used as a dynamo for power absorption or as a motor for measuring friction losses.

The engine is coupled to the armature shaft while the field magnets are supported on ball bearings so that the whole field magnet frame is capable of free rotation through a small arc.

The torque produced by rotating the armature in the magnetic field of the field magnets tends to make the field magnet frame rotate with the armature.

This rotation is prevented by a steel-yard balance arrangement which allows the field magnet frame to be kept "floating" and allows of very accurate measurement of the torque and power developed.

By the introduction of suitable resistances a very wide range of torque and speed may be investigated.

The calculations are precisely the same as with the Prony brake previously described.

Indicated Horse-power. — Since it is necessary to find the work done in foot-pounds per minute in order to find the horse-power of an engine, it is necessary to find the mean force acting on the piston and multiply this by the distance moved through per minute. The mean force is found by taking an indicator card from the engine cylinder. This indicator card is nothing more nor less than a graphical record of the pressure in the cylinder at all points of the cycle, but, knowing the laws of gases, many interesting facts may be read "between the lines."

The ordinates of the indicator card represent unit pressures on the piston when the length of the card represents the volume displaced per stroke of the engine or the ordinates represent total pressures on the piston when the length of the card represents the stroke of the engine in feet. Thus in either case the area of the indicator card represents the work done in foot-pounds per working stroke. If the mean total pressure on the piston is computed by multiplying the mean height of the card in pounds per square inch P by the area of the piston in square inches A and if this total pressure is multiplied by the length of the stroke in feet L the result will be the foot-pounds of work done in the cylinder per cycle since there is one working stroke for each cycle. Since the horse-power of any engine is the foot-pounds of work done per minute divided by 33,000, the general horse-power formula may be written

$$\text{I.H.P.} = \frac{PLAN'}{33{,}000} \qquad (1)$$

which is true for any type of gas or steam engine where N' is the number of cycles per minute. In a double-acting, single-cylinder steam engine there are two cycles per revolution or 2 N cycles per minute where N is the number of revolutions per minute. The horse-power formula for that engine then becomes

$$\text{I.H.P.} = \frac{2\,PLAN}{33{,}000}. \qquad (2)$$

For a two-cycle, double-acting, single-cylinder gas engine the same formula would apply since there are two cycles per revolution as in the steam engine. If the engine is a four-cycle, double-acting, single-cylinder gas engine formula (1) becomes

$$\text{I.H.P.} = \frac{PLAN}{33{,}000}, \tag{3}$$

since there is a cycle for each revolution.

Investigation of formulas (1), (2) and (3) will show that to change (1) into a form involving N, the revolutions per minute for any type of engine, it is only necessary to multiply (1) by the number of cycles per revolution and substitute N for N'. Thus in a four-cycle, single-acting, single-cylinder gas engine there are two revolutions per cycle or $\frac{1}{2}$ cycle per revolution and the horse-power is

$$\text{I.H.P.} = \frac{PLAN}{2 \times 33{,}000}. \tag{4}$$

This suggests a general formula for use with single-acting four-cycle engines with any number of cylinders, n,

$$\text{I.H.P.} = \frac{n\,PLAN}{2 \times 33{,}000}. \tag{5}$$

Example: As a practical example in the use of the above formula, the author quotes here from a test made by him on a two-cylinder, double-acting, two-cycle engine. The motor cylinders had a diameter of $24\frac{3}{4}$ inches and a stroke of $43\frac{3}{8}$ inches. The average mean effective pressure for the four combustion chambers was 62 pounds per square inch. The net area of piston, allowing for rods, was 453 square inches and the revolutions per minute 103.23.

The horse-power of the engine then becomes

$$\text{I.H.P.} = \frac{4 \times 62 \times 43.375 \times 453 \times 103.23}{33{,}000 \times 12} = 1270.$$

This indicated horse-power included, in this two-cycle engine, the work done by the gas and air pumps, which amounted to about 262 indicated horse-power, leaving a net indicated horse-power of 1018. The output of the generator was 600 kilowatts, kept constant by means of a water rheostat. With a generator efficiency of 93 per cent, the brake horse-power of the engine would be

$$\frac{600}{.93 \times 0.746} = 864.$$

The mechanical efficiency of this engine was then

$$\frac{864}{1270} = 0.68 \text{ or } 68 \text{ per cent.}$$

Thermal Efficiency. — The mechanical efficiency of a gas engine has been explained and illustrated by the above example. There is still another efficiency which is of more importance than the mechanical efficiency, the **thermal,** or heat efficiency. In order to understand this term we must first look at the relation of heat and work. The unit of heat, **British thermal unit,** has already been defined as the amount of heat required to raise the temperature of 1 pound of water from 62° to 63° F. Since heat can be transformed into work and work into heat, there must be a known relation existing between the two. That relation is called the mechanical equivalent of heat, and is 778 foot-pounds per B.t.u. This is usually denoted by *J*.

In a perfect engine, then, for every B.t.u. put into the engine we should take out 778 foot-pounds of work. Since there are 33,000 foot-pounds per minute in a horse-power, the heat to be supplied, theoretically, per horse-power, per minute is

$$\frac{33{,}000}{778} = 42.42.$$

Another value that is used and referred to frequently is the heat per horse-power hour, which is $60 \times 42.42 = 2545$.

Since the efficiency of a gas engine is never perfect, or equal to 1, that is since we can never get 778 foot-pounds of work for each B.t.u. put in, we must put in more than 2545 B.t.u. in order to take out one horse-power for a period of one hour.

The general expression for any efficiency is

$$\frac{\text{Output}}{\text{Input}} = \text{efficiency}.$$

In any engine, then, to find the thermal efficiency, it is only necessary to find the heat actually supplied per horse-power and divide it into the output, **per horse-power hour,** which is always 2545 B.t.u.

The heat input must be found by measuring the fuel delivered to the engine per horse-power per hour, and multiplying this value by the heating value per unit of the fuel. The fuel may be based on the indicated horse-power or the brake horse-power, giving us two values of thermal efficiency. The efficiency based on the indicated horse-power is used frequently in design but that based on the brake horse-power, or output, is the one that tells the story of an economical or a wasteful engine. A purchaser of an engine

is interested in knowing how much power he is going to have available for actual use, per ton of coal or per 1000 cubic feet of gas as the case may be. He does not care what percentage of heat will be transformed into work in the engine cylinder but what percentage will be transformed into work at the switchboard or at the belt drive.

The ratio between thermal efficiency based on brake horse-power and that based on indicated horse-power is the mechanical efficiency, or

$$\frac{\text{Efficiency based on B.H.P.}}{\text{Efficiency based on I.H.P.}} = \text{mechanical efficiency.}$$

To illustrate the method of finding the efficiency of a gas engine, the author gives here the results of the gas engine test referred to above. The fuel was blast-furnace gas, measured by means of a Venturi meter. It was found that the consumption of gas per hour during the 600-kilowatt test was 129,510 cubic feet at standard conditions. Since the indicated horse-power was 1270, the fuel consumption per I.H.P. hour was 102 cubic feet. The heating value of the gas at standard conditions was 94.59 B.t.u. per cubic foot. The efficiency based on indicated horse-power was then

$$\frac{2545}{102 \times 94.59} = 0.264 \text{ or } 26.4 \text{ per cent.}$$

Since the brake horse-power was 864, the fuel consumption was 150 cubic feet per brake horse-power hour, giving an efficiency based on brake horse-power of

$$\frac{2545}{150 \times 94.59} = 0.1793 \text{ or } 17.93 \text{ per cent.}$$

It will be noted that $\frac{0.1793}{0.264} = 0.68$, the mechanical efficiency found above.

Efficiency at Various Compressions.—It was brought out in Chapter III that the efficiency of the Otto cycle increased with compression according to the formula $E = 1 - \frac{1}{r^{(n-1)}}$ where r = compression ratio.

The history of the development of the internal combustion engine may be expressed as an effort to increase compression.

Unfortunately many practical difficulties arise owing to some gases being liable to detonation owing to their peculiar chemical composition.

For this reason exhaust gas is used as a diluent in some large gas engines because of its effect in preventing detonation whilst alcohol functions in the same way when mixed with benzole and gasoline in automobile engines.

Quite recently anti-knock liquids have been produced which allow of compression ratios up to 14 : 1 being used even with the most detonatory type of gasoline or even kerosene. Thus the detonation limit to compression ratio is definitely under control. The problem is now that of mechanical design to enable high compression ratios to be used.

Air Standard Efficiency.—Fig. 144 shows the relation between compression ratio and efficiency upon the assumption that the working substance is air with $n = 1.41$ and that no heat is lost to cylinder walls and no change in specific heat occurs with temperature. This forms a standard of comparison with engines working upon the Otto cycle and represents the absolute maximum obtainable for any given compression ratio. The deviation of the practical case from the air standard is accounted for as follows:

First and foremost is that due to the fact that the temperature of the mixture after combustion and before expansion is very much less than that calculated from a knowledge of the mass of the mixture, its specific heat and heat supplied. This loss is the subject of much discussion and may be conveniently referred to as a loss due to change of apparent specific heat. It is probably due in part to dissociation and to direct luminous radiation into the cylinder walls.

However this may be, the fact remains that some 16 per cent of the heat value of the mixture disappears in this manner. A further 12 per cent is given up to the cylinder cooling water via the cylinder walls while some 7 per cent is lost due to imperfect combustion and early opening of the exhaust valve, *i.e.*, the rounded corners of the actual indicator diagram as compared to the ideal. We have, therefore, a necessary and unavoidable loss of some 33 per cent of the air cycle efficiency. The relation between the efficiency after these deductions or, as it is called, the indicated efficiency to the air cycle efficiency is called the

"Relative Efficiency" of the engine a term of the utmost importance in the study of gas engine literature. As stated the value attainable under conditions of first-class design and con-

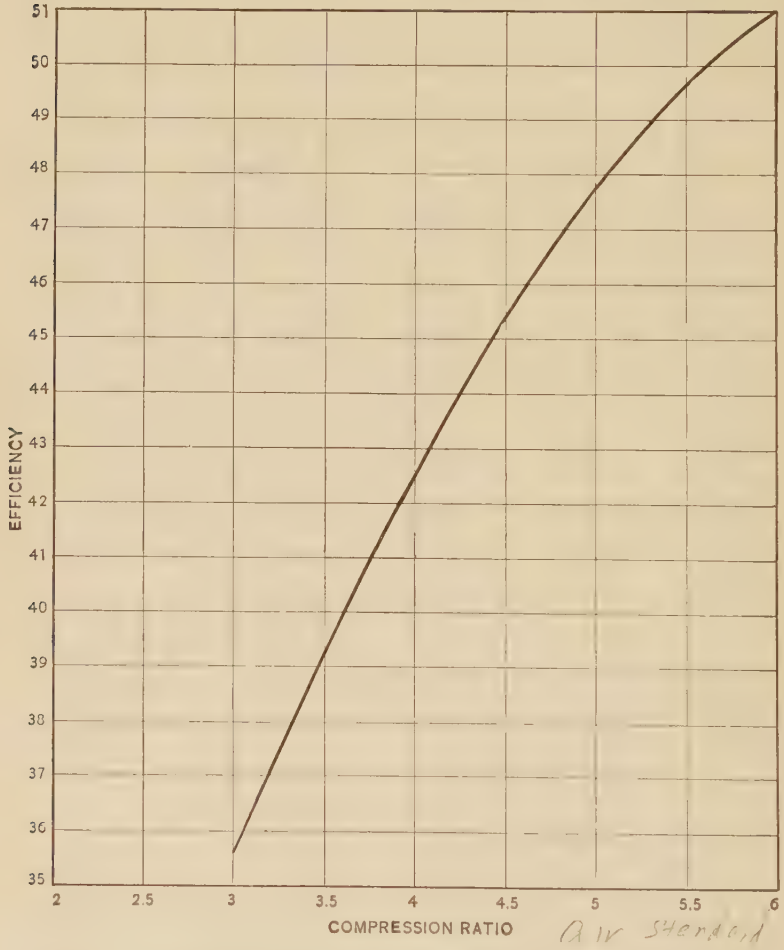


FIG. 144.—Relation between Compression Ratio and Efficiency.

struction is some 66 per cent. An indifferent engine will fall as low as 50 per cent while a super engine may attain 68 to 69 per cent.

In general the relative efficiency is found to vary with the engine speed falling off due to the greater time element in cooling

and lower turbulence at low speeds and at high speeds due to wire drawing with greater exhaust gas dilution and less effective combustion.

Knowing the relative efficiency and the mixture strength in foot-pounds per cubic inch the indicated mean effective pressure may be calculated providing that the amount of mixture taken into the cylinder per stroke is known.

This depends upon the volumetric efficiency of the engine or the relation between the weight of charge in the cylinder at the instant of closing the inlet valve to that of a charge of equal volume but at atmospheric temperatures and pressure. This depends upon

1. The degree to which the charge is throttled by its passage through the carburetor inlet manifold and valves.

2. The extent to which it is heated during its journey from the carburetor to the cylinder.

Turbulence is the state of violent agitation which must exist in the mixture at the moment of ignition. If this is deficient, *i.e.*, if the charge is more or less quiescent then combustion only takes place slowly and power and efficiency are enormously affected so much so that in some cases over-valved engines have not been able to run at all until this defect has been corrected. The designer is, therefore, confronted with another of the many dilemmas of design. To maintain volumetric efficiency adequate valve openings must be employed while if these are too large turbulence is reduced and combustion efficiency with it.

Another very serious factor is that large valves call for a combustion chamber of extended area for their accommodation so that the resultant compression space when the piston is at the end of the compression stroke is far from compact.

Proportions of Cylinders.—The ratio of the length of stroke to diameter of cylinder is an arbitrary one, and a little investigation will show that this ratio may vary from 1 to 2. As a rule the higher the engine speed, the nearer to 1 should be the ratio. Recently the ratio has been increased in this line of work until in some engines it is as high as 1.5.

The condition which influences this ratio more than anything else is the shape of the combustion chamber. The most effective and efficient combustion chamber is one in which the ratio of enclosing area to volume is small. If the combustion chamber is

cylindrical and the length equal to the diameter, then the ratio of area to volume is 6. If the length be equal to $\frac{1}{2}$ the diameter, then the ratio is 8, but if the length be equal to $\frac{1}{4}$ the diameter, the ratio will be 12. If we assume the combustion chamber volume, or, what is the same thing, the clearance volume, to be 25 per cent, corresponding to about 100 pounds compression, the stroke of the engine will have to be four times the diameter of the piston to give a length of clearance volume equal to the diameter. This ratio of stroke to diameter is absurd, of course. If the ratio of length of clearance to diameter is reduced to $\frac{1}{2}$, then the stroke of the engine will be twice the cylinder diameter, a ratio that might be taken as the maximum.

Working from the ratio of stroke to diameter we may find the value of the ratio of length of combustion chamber to diameter. Assuming a ratio of stroke to diameter of 1.5, we see that with 25 per cent clearance, the length of combustion chamber would be $\frac{3}{8}$ of the diameter. Some engines are built with a ratio of length to diameter of combustion chamber of $\frac{1}{3}$ without considering the ratio of stroke to diameter.

In double-acting engines the combustion space is so irregular in form that any assumption of ratio as above is of little value. In high-compression engines, such as the Diesel and semi-Diesel type, the combustion space is often made of smaller diameter than the cylinder proper, in order to get good proportions and at the same time have space for the insertion of valves.

Speed of Engines.—The speed of gas engines, that is, the revolutions per minute, is governed by the character of the work to be done or by the piston speed. Automobile motors run at extremely high speeds in order to secure the maximum power per unit of weight. The maximum speed of such motors may be taken at 3200 revolutions per minute. A five-inch stroke engine at this speed would have a piston speed of 2666 feet per minute. The rated horse-power of a four-cylinder, four-cycle automobile engine with six-inch cylinders would be

$$\text{H.P.} = \frac{6^2 \times 4}{2.5} = 57.5.$$

Since this formula,

$$\text{H.P.} = \frac{D^2 \times \text{no. cylinders}}{2.5},$$

is based on 1000 feet piston speed, an engine with a six-inch diameter cylinder and a six-inch stroke at 1800 revolutions per minute might be rated by the manufacturer at $1.8 \times 57.5 = 103.5$ horse-power.

The piston speed of stationary engines varies from 400 up to 1000 feet per minute. The gas engine is inherently a high-speed engine on account of rapid combustion of the gases, but high speed means rapid wear, and commercially it is better to strike a happy medium around 600 to 800 feet per minute.

Marine engines should be designed to give proper propeller speeds. The maximum efficiency of modern propellers is around 100 revolutions per minute, but there are cases where they are run much higher with a corresponding loss in efficiency. They are seldom run at lower speeds.

Scavenging. — One disadvantage in running a gas engine at high speed is the effect on the scavenging which is the process of cleansing the cylinder of burned gases. High piston speed means high velocity of gas through the valves, which in turn results in great fluid friction. In a four-cycle engine rarefied suction charge and high exhaust pressures will be the outcome, both tending to reduce the mean effective pressure, so that the increase in horse-power is not proportional to speed.

In two-cycle engines the effect of high speed is still more injurious. When scavenging is done by the crank-case compression, high speed necessitates larger passages from the crank-case to the cylinder than can be conveniently used. If mechanically operated scavenging valves are used, these valves must be large in order to get the proper amount of scavenging air or mixture through them in the short time available. Since on a two-cycle engine these valves must be opened and closed every revolution, high speed is prohibitive on account of trouble with the heavy valve gear at such speeds. Thus, as far as the valve gear is concerned, a four-cycle engine may operate at twice the speed of a two-cycle.

Perfect scavenging in a two-cycle engine is seldom accomplished. The best results are obtained when the scavenging air or mixture enters the cylinder at low pressure, not more than 6 pounds above the atmosphere, and in such a way that no eddy currents are formed. The entering air should drive the burned charge ahead of it in a path as straight as it is possible to make it. Any turn in the current of air, or any protuberance in the cylinder, causes the

air and burned charge to mix and defeats the purpose of the scavenging air. For this reason, mechanically operated valves in the head of the cylinder are more efficient than scavenging ports uncovered by the piston.

EXERCISES

1. (*a*) Find the b.h.p. of an engine which has an i.h.p. of 47.3, efficiency 81 per cent. (*b*) Find the i.h.p. of an engine with an efficiency of 86 per cent, b.h.p. 302.5.

2. On a prony brake test the total load on the scales was 528 pounds, radius of brake 4 feet, speed 275 r.p.m. Find the b.h.p. if the weight on the scales with the band loose was 45 pounds when the wheel was turned forward and 25 pounds when the wheel was turned in the opposite direction.

3. Find the i.h.p. of a double-acting, single-tandem, four-cycle gas engine, 28 × 40 inches, 115 r.p.m. M.e.p. = 73.5 lbs. per sq. in.

4. Find the i.h.p. of a two-cycle, single-acting, single-cylinder engine, 7 by 11 inches running 325 r.p.m. M.e.p. = 67.8 lbs. per sq. in.

5. A test made on a gas engine using natural gas as fuel showed that when the i.h.p. was 76 the gas consumption was 859 cu. ft. per hour, heating value 935 B.t.u. per cu. ft. Find the thermal efficiency based on i.h.p. and on b.h.p. if the mechanical efficiency is 85 per cent.

6. Find the horse-power of an eight-cylinder automobile engine with cylinders 3 in. dia., 5 in. stroke. (*a*) At 1000 ft. piston speed. (*b*) At 2500 r.p.m.

CHAPTER XVIII

DESIGN OF THE GAS ENGINE

Calculation of Cylinder Volumes — Allowable Compression — Construction of Indicator Cards — Calculations for Diesel Engines

General. — In any problem of machine design, there is no fixed rule as to the course to pursue; that is, there is no exact method of procedure to follow. There are numerous ways in which a design problem may be worked out, and possibly several of them will appear to be about equal logically. Probably the best course to take is the one involving the fewest assumptions. In any case some assumptions must be made. Some parts must necessarily be designed and redesigned and then designed again in order to fit in with other parts. The beginner in design work does not know just which path will involve the fewest assumptions, and it is only with experience that the faculty for working out a long problem logically can be attained.

The parts of a machine support loads or transmit forces in doing the useful work for which they are designed. They are proportioned to take these loads without breaking and at the same time without an uneconomical expenditure of material. The parts, then, must be strong and rigid. In general, the stronger the material, the less the bulk and consequently the cost. For instance, some bridges have been built of nickel steel instead of carbon steel, even though the nickel steel costs considerably more per pound than the more common kind. The saving was due to the fact that the allowable stress of the nickel steel is much higher than of carbon steel. This reduced the dead weight of the bridge since smaller members could be used. Reducing the dead weight decreased the stresses and permitted a further reduction in the size of the members and consequently in the weight and cost of the bridge.

In starting any design problem, there are four points or milestones, as it were, where halts should be made long enough for

reflection. These are: First, **analyze;** second, **theorize;** third, **modify;** fourth, **construct.**

The Analysis. — Any problem in design, no matter how small, is important enough to demand careful analysis on the part of the designer. Too little time, rather than too much, is spent on this very important beginning. The success of the whole design rests on the analysis and just as this analysis is accurate, so will the results of the work be of value in proportion. Among the important things to be considered in the analysis are the loads on the part under consideration, how these loads are applied, their magnitudes and directions. The unusual conditions or loads which are likely to come into action during the life of the machine must be taken care of, and, in many cases, it is the unusual that causes trouble.

The Theory. — After the analysis has been made, the conditions that exist must be handled, first, according to theory. Sizes of parts may generally be determined from purely theoretical principles. If several conditions exist at the same time, they must be properly combined by theory. And, as in the analysis, the theory must not be at fault. That is, a bending load must not be treated as a load in straight tension; proper stresses must be used with a given material; loads which the analysis has shown to be suddenly applied must not be treated as dead loads, etc.

Modification. — After the calculations have been made from a theoretical standpoint, they must be changed to suit practical considerations. It is very easy for an engineer to design a machine in the drafting room; it is an entirely different matter to construct this machine in the shop and do it in an economical manner. Parts made of cast iron require careful consideration in order to make it possible to construct the pattern and mold the piece without excessive cost. Where many parts are to be made identical, they should be modified to allow the use of automatic machinery or jigs. Standard parts should be used when they are available and when they will answer the purpose as well as a specially constructed part.

Construction. — After the modifications have been made, the construction of the drawing may be carried out. During the preliminary and final calculations, free-hand sketches should be made and dimensions attached as they are determined. These

sketches are valuable helps in detecting errors and in making the actual drawing. This drawing should be simple; it should show everything that it is necessary to show and no more. Dimensions should be used freely and should be put on in plain figures. Nothing should be left to the imagination of the mechanic who is to do the actual construction of the part — possibly he will have no imagination.

The Gas Engine Design Problem.—The gas engine design problem does not differ from any other in that certain conditions must be fixed in order to allow a logical treatment of the subject. We must know what type of engine is to be built, the size and the fuel available. It is also necessary to know, in many cases, the use to which the engine is to be put. We might not design an engine for driving an electric generator just as we would one that is to be belted to a line shaft. The method to be used in this book makes it necessary to know also the back pressure and suction pressure in the cylinder. These two values must be assumed. Further along we will see that other assumptions must also be made.

In order to help the reader in the work that follows, a brief outline is given here of the method of procedure in this problem. The first step is to make calculations that will enable us to find the size of the cylinder. This work has been explained in a previous chapter. After the size of the cylinder has been determined the mean effective pressure may be computed and the indicator card drawn. Since the area of the piston and the pressure on it are known at this point, the load on the engine, that is, the load for which the parts are to be designed, is known, and the design of these parts may begin. The reciprocating parts, piston, rod, cross-head and connecting-rod, are designed first. The weights are found, inertia curves drawn, and these curves combined with the original indicator card giving a curve of net or actual pressures on the piston. These pressures are then transferred to the crank-pin, giving a "rotative effort diagram" from which the weight of the flywheel necessary for the given type of engine is found.

Knowing the weight of the flywheel and the thrust along the piston- and connecting-rods, the shaft may be designed. The cylinder thickness may be determined and the diameter of valves found. After determining the size of the valves, the valve gear-

ing is designed, together with the governor, ignition mechanism and other auxiliaries.

In order to make certain steps and calculations as clear as possible, an example will be worked from time to time to illustrate points in the work as it is developed. For this example we will assume a four-cycle, double-acting engine of 225 brake horsepower, operating at 140 r.p.m. Fuel to be producer gas which has the following composition: H_2 = 14.9 per cent, CO = 19.7 per cent, CO_2 = 10.0 per cent, CH_4 = 3.6 per cent and N_2 = 51.8 per cent, by volume.

TABLE XI

ALLOWABLE COMPRESSIONS FOR VARIOUS FUELS

Fuel.	Compression pressure, lbs. gauge.	Approximate clearance, per cent piston displacement.
Gasoline: Automobile motors, high-speed	60–90 Aver. 75	32
Stationary slow-speed engines	70–90 Aver. 80	30
Kerosene: Hot-bulb ignition, fuel injected	30–75	35–40
Previously vaporized	45–85 Aver. 65	35
Natural gas	115	22
Producer gas	100–160	26 to 15
Producer gas, water-cooled pistons	135	20
Illuminating gas	60–100 Aver. 80	26
Blast-furnace gas	120–190 Aver. 155	17

Table XI shows the compression allowable for various fuels from which it can be shown that the appropriate compression ratio for producer gas is 6.2:1.

The desirability of reckoning compression in terms of the ratio of maximum cylinder volume to cylinder volume at the end of the compression stroke should be emphasized. Compression pressure is very difficult to measure, on the other hand the ratio of volumes is a definite geometrical problem which can be determined quite accurately.

The air standard efficiency corresponding to a compression ratio of 6.2:1 is about 55 per cent. The relative efficiency varies, as has been stated, with the general design of the engine.

The method of computing the compression ratio for an assumed compression pressure will be shown in the discussion of the indicator card further on in this chapter.

Table XII indicates reasonable value for engines various types.

TABLE XII

Types of engines	Valve arrangement	Relative efficiency, per cent
High-speed engines	T head	52
High-speed engines	L head	55
High-speed engines	Valves in head	62–68
Producer gas engine	Valves in head	65–70
Natural gas	Valves in head	62–70
Diesel engine (22″ × 29.5″)	Valves in head	61

Defective design in respect of ignition or valve gear may affect these figures very considerably. The figure of 70 per cent for producer gas is one of the highest ever recorded so that in the interests of practice and prudence it may be well to take this at 65 per cent. We have then for the above example:

		PER CENT
	Air standard efficiency	= 55.0
	Relative efficiency	= 65.0
or	Indicated thermal efficiency .55 × .65	= 35.8

From Table II the theoretical air per cubic foot of gas is found to be 1.176 cu. ft. From Table I the heating value is found to be 149 B.t.u.'s per cu. ft. The number of B.t.u.'s per cu. ft. of mixture is, therefore, $\frac{149}{2.176} = 69$. The number of B.t.u.'s of this mixture actually realizable if the gases were at normal temperature and pressure at the end of the suction stroke would be 69 × 35.8 per cent = 24.6 B.t.u.'s per cu. ft.

Given that the valve gear is properly designed so that the gas velocity through the inlet valves is about 160 ft. per sec. and that there are no undue restrictions in pipe work, the volumetric efficiency will be about 78 per cent, *i.e.*, 96 per cent of the maximum possible corresponding to a pressure at the end of the suc-

tion stroke of $\frac{14.7 \times 96}{100} = 14$ lbs. per sq. in. and a temperature of 147° F.

The result of this rarefaction due to the combined effects of heat and throttling is, therefore, to reduce the available heat value of 1 cu. ft. of mixture at normal temperature and pressure to $\frac{24.6 \times 78}{100} = 19.3$ B.t.u.'s per cu. ft. 19.3 B.t.u.'s per cu. ft. is obviously 19.3×778 ft.-lbs. per cu. ft.—778 being Joule's equivalent or $\frac{19.3 \times 778}{1728}$ ft.-lbs. per cu. in. = 8.8 ft.-lbs. per cu. in.

The next step is simple but should be followed very carefully.

An engine of 1-inch area, 1-inch stroke using this mixture would develop 8.8 lbs. of work per stroke or 12×8.8 in.-lbs. per stroke = 105 in.-lbs. per sq. in. and the average pressure during the stroke of 1 in. would be 105 lbs. per sq. in.

This is the indicated mean effective pressure developed under the above conditions of mixture strength, air standard, relative and volumetric efficiencies. Knowing this the remainder is simple arithmetic.

The horse-power required is 225 actual. Assuming a mechanical efficiency of 85 per cent and an overload capacity of 15 per cent the maximum indicated horse-power required is $\frac{225 \times 115}{85} = 302$.

The specified engine speed is 140 r.p.m. and as the engine is double acting there will be a suction stroke each revolution.

$$\text{Indicated horse-power} = \frac{\text{PLAN}}{33{,}000 \times 12}$$

where P = Indicated m.e.p. = 105
L = Stroke in inches (required)
A = Area of piston in inches (required)
N = Number of impulses = 140

$\therefore$ we have whence $302 = \frac{105 \times L \times A \times 140}{12 \times 33{,}000}$

or $LA = 8200$ cu. in. or 4.75 cu. ft.

The ratio of length to diameter of cylinder bore is arbitrarily fixed by practical considerations. As the lower limit, we might say that it is never made less than one, and for the upper limit

we may safely assume two to be the value. The considerations that fix the value between these limits are speed, available space and the shape of the combustion chamber. The tendency in high-speed engines is to make the ratio small, that is, nearer the lower limit than the upper. For a given suction displacement and a given speed there is an infinite number of ratios that could be used. However, the smaller the ratio, the lower will be the piston speed, and it is important to keep down the piston speed in stationary engines.

The space occupied is important in automobile and marine engines. If large values be assumed for this ratio for these types of engines, the result will be high machines of the vertical type that will be top heavy. By making the diameter small and stroke long, the crank radius and connecting-rod length are increased, both helping to increase the height of the unit. This question of long or high units is not so important in stationary engines, and ratios may be made greater in that type.

The shape of combustion chamber is important, particularly in high-compression engines. If the combustion chamber is cylindrical and of the same diameter as the cylinder, the greater the ratio of $\frac{L}{d}$ the longer will be the combustion chamber in proportion to the diameter. If the length of the combustion chamber h is large compared with the diameter, say equal to the diameter, the surrounding area is a minimum in proportion to the volume. The ratio of the enclosed area to the volume of the combustion chamber does not change much until h becomes less than one-third of the cylinder.

In Fig. 145 are shown the different ratios of $\frac{h}{d}$ for two values of $\frac{L}{d}$ and compressions of 110 and 500 pounds gauge. In A the proportion of $\frac{h}{d}$ is not too small to be prohibitive, while that shown in B is larger than is necessary. A value of $\frac{h}{d}$ between these two, say of $\frac{1}{3}$, would be preferable to either A or B. In C we have the ratio of $\frac{h}{d}$ that would exist in a Diesel engine if $\frac{L}{d} = 1$, and in D the ratio if $\frac{L}{d} = 2$. In both C and D the ratio of $\frac{h}{d}$ is too

small for good results. To remedy this, the diameter of the combustion chamber should be made smaller than the cylinder diameter.

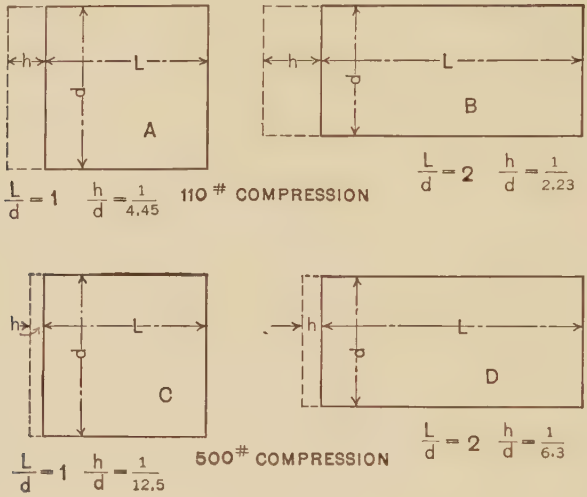


FIG. 145.

From the foregoing it will be seen that a value of $\frac{L}{d}$ of 1.5 will give good proportions in the combustion chamber for a 6.2:1 compression ratio. Using these proportions we have

$$\frac{\pi d^2}{4} \times 1.5d = 4.75 \text{ cu. ft.}$$

whence $$d = \sqrt[3]{\frac{4.75 \times 4}{\pi \times 1.5}} = 1.59 \text{ ft. or } 19.2 \text{ in.}$$

and $$L = 19.2 \times 1.5 = 28.8 \text{ in.}$$

Generally if

H = Heat value of fuel in ft.-lbs. per cu. in.
E = Air standard efficiency
R_E = Relative efficiency
V_e = Volumetric efficiency
M_E = Mechanical efficiency
N = Number of impulses per minute
V = Cylinder volume in cu. in.
D = Cylinder diameter ft.

$$S = \frac{S}{D} = R$$

$$S = \text{Stroke in ft}$$

$$\text{Indicated m.e.p.} = H \times 12 \times E \times R_E \times V_e$$

$$\text{Brake m.e.p.} = \text{Indicated m.e.p.} \times M_E$$

$$\text{I.H.P.} = \frac{\text{Indicated m.e.p.} \times V \times N}{33{,}000 \times 12}$$

$$\text{B.H.P.} = \text{Indicated h.p.} \times M_E$$

$$V = \frac{33{,}000 \times 12 \times \text{b.h.p.}}{\text{Brake m.e.p.} \times N \text{ cu. ft.}}$$

$$\text{Cylinder bore} - D = \sqrt[3]{\frac{V \times 4}{\pi \times R}}$$

$$\text{Stroke} = D \times R$$

As has been stated the above method is perfectly general in its application to all engines working on the Otto cycle. Certain slight modifications are sometimes needed as between one fuel and another, notably the correction for contraction of volume after expansion in the case of artificial gas and the increase in volume after explosion in the case of rich gasoline mixtures. The maximum values of these effects are 3 per cent and 8 per cent respectively. Such corrections, however, are small and do not effect the general fundamental principles involved.

The Oil Engine.—To further illustrate the principles discussed above in determining cylinder sizes, the case will be taken of a single-cylinder, single-acting Diesel engine required to deliver 50 b.h.p. at 225 r.p.m. with an overload capacity of 15 per cent. The composition of the fuel will be taken as H_2, 14 per cent; carbon, 84 per cent; balance, 2 per cent by weight.

From Table 11 the air required by per pound is

Carbon $152 \times .84 = 128$

Hydrogen $456 \times .14 = 64$

$= 192$ cu. ft. at 62° F.

The heating value of the oil is found from the equation given in Chapter VI

$$\text{B.t.u.} = 14{,}500\,C + 52{,}230\left(H - \frac{O}{8}\right)$$

$$= 14{,}500 \times 0.84 + 52{,}230 \times 0.14$$

$$= 19{,}480$$

$\therefore$ the number of B.t.u.'s per cu. ft. of mixture for complete combustion is $\frac{19{,}480}{192} = 101.$

$$\text{Number of ft.-lbs. per cu. in.} = \frac{101 \times 778}{1728} = 45$$

Allowing 50 per cent excess air this becomes 30 ft.-lbs. per cu. in.

It is characteristic of the Diesel engine that much weaker mixtures can be used than with the Otto cycle thus keeping cylinder temperatures and promoting volumetric efficiency.

In Diesel engine practice generally the compression pressure is about 500 lbs. per sq. in. corresponding to a compression ratio 12.24:1.

Proceeding as in the case of the producer gas engine we have

H = Heat value of fuel in ft.-lbs. per cu. in. = 30
E = Air standard efficiency (r = 12.2 : 1) = 64
R_E = Relative efficiency = 61.0 per cent
V_E = Volumetric efficiency = 78 per cent
M_E = Mechanical efficiency = 80 per cent
N = Number of impulses per minute = 112.5
D = Cylinder diameter in ft. required
L/D = Stroke bore ratio = 1.5

Then Indicated m.e.p. $= H \times 12 \times E \times R_E \times V_E$
$= 30 \times 12 \times .64 \times .61 \times .78$
$= 91$ lbs. per sq. in.

Brake m.e.p. = Indicated m.e.p. $\times M_E$
$= 91 \times .80$
$= 72.8$ lbs. per sq. in.

For 15 per cent overload maximum b.h.p. $= 50 \times 1.15$
$= 57.5$

$$V = \frac{33{,}000 \times 12 \times \text{b.h.p.}}{\text{Brake m.e.p.} \times N}$$
$$= \frac{33{,}000 \times 12 \times 57.5}{72.8 \times 112.5 \text{ cu. in}}$$
$= 2{,}850$ cu. in.
$= 1.65$ cu. ft.

Cylinder diameter $= D = \sqrt[3]{\frac{4 \times V}{\pi \times 1.5}}$
$= \sqrt[3]{\frac{4 \times 1.65}{3.14 \times 1.5}}$
$= 1.12$ ft.
$= 13.5$ in.

Cylinder stroke $= 1.5 \times 13.5$ in.
$= 20.5$ in.

The Automobile Engine.—In the case of an automobile engine it is usual to decide the cylinder dimensions first and calculate the horse-power afterwards.

This is, of course, irrational from the strictly engineering point of view but not more so than numerous other items in automobile construction.

In brief the public associates a certain performance with a certain size of engine and it is in practice a competition between designers to get the maximum out of a given cylinder rather than to assume a desired performance and design the cylinder accordingly.

For an automobile engine with cylinders of 3-in., bore, 5-in. stroke, compression ratio 4.8:1, running on standard gasoline at a speed of 1600 r.p.m. the horse-power which may be developed per cylinder is determined as follows:

Mixture Strength.—One pound of gasoline requires 15 lb. air for complete combustion. One pound of air at normal temperature and pressure occupies 13 cu. ft. $\therefore$ 15 lb. air occupies $15 \times 13 = 195$ cu. ft. Calorific value of 1 lb. gasoline = 19,120 B.t.u.'s $\therefore$ Mixture strength of standard mixture $= \frac{19{,}120}{195}$ or 100 B.t.u. per cu. ft. $= 100 \times 778$ ft.-lb. per cu. ft. or $\frac{100 \times 778}{1728} = 45$ ft.-lb. per cu. in.

Air Cycle Efficiency.—From Fig. 144 the air cycle efficiency for a compression ratio of 4.8:1 = 46.8 per cent.

Relative efficiency varies with shape of combustion chamber and the average values

	Per Cent
T-head engine	52–55
L-head engine	57–56
Overhead-valve engine	57–69

Assuming the engine in question is a conventional L-head touring-car engine the relative efficiency may be taken as 55 per cent.

Volumetric Efficiency.—The ideal volumetric efficiency has been shown to be 81 per cent assuming that the gas velocity through the valves at 1800 r.p.m. is 150 ft. per sec. and making allowance for throttling in the carburetor and inlet manifold and the necessary heating of the charge to insure vaporization the volumetric efficiency in practice will be about 75 per cent.

The energy realizable in the form of work done by the gas upon the piston is then equal to

Mixture strength $\times$ air cycle efficiency $\times$ relative efficiency $\times$ volumetric efficiency

$= 45$ ft.-lb. per cu. in. $\times .468 \times .55 \times .75$
$= 8.66$ ft.-lb. per cu. in.
$\therefore$ Indicated m.e.p. $= 8.66 \times 12$
$= 104$ lb. per sq. in.

Brake mean effective pressure = Indicated m.e.p. $\times$ mechanical efficiency
$= 104 \times .85$
$= 88.5$ lb. per sq. in.

From this the brake horse-power is easily determined by the standard formula

Where P = Brake mean effective pressure
L = Stroke in feet

$$\text{B.H.P.} = \frac{PLAN}{33{,}000}$$

A = Piston area in sq. in.
N = Number of explosions

$$= \frac{86 \times 5 \times 3^2 \times .7854 \times 1800}{12 \times 33{,}000 \times 2}$$

$= 7.2$ brake horse-power per cylinder.

The brake horse-power developed in a four-cylinder engine at 1800 r.p.m. designed as above would, therefore, be 28.8, a six-cylinder engine $28.8 \div 1.5 = 43.2$ and so on.

In applying these figures over a range of engine sizes and speeds considerable judgment is required.

In the case taken of an engine with cylinders 3×5 in. the volumetric efficiency at 3000 r.p.m. with a gas velocity of 220 ft. per sec. through the inlet valve would probably not exceed 66 per cent while the relative efficiency at 3000 r.p.m. would probably fall to some 50 per cent.

The horse-power, therefore, does not rise in direct proportion to the speed but attains a peak somewhere between 2500 and 3000 r.p.m. in well designed engines and then begins to fall.

Similarly at very low speeds the effects of cylinder cooling, bad distribution of the working mixture with consequent defec-

tive combustion cause the horse-power to fall off in greater proportion than the reduction in speed.

The mechanical efficiency or friction losses would be affected as indicated in the table previously given.

These latter are greatly dependent upon the weight of their reciprocating parts, *i.e.*, the piston and connecting-rod. For both these parts aluminum alloys are now widely used and have far reaching effects not only in the reduction of internal friction through reducing the bearing pressures set up by inertia but because they allow bearing areas to be reduced by some 30 to

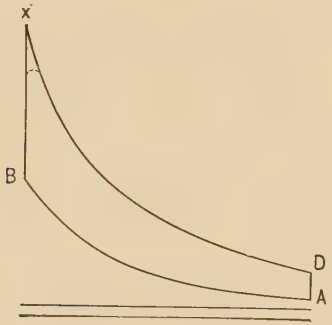


FIG. 146.

40 per cent, this making it possible for an engine to be shortened in overall length.

The Indicator Card.—The cylinder dimensions having been determined, it is desirable, in the case of stationary engines, at least, to construct the indicator card such as the one shown in Fig. 146. The point *B*, the pressure at the end of compression, has been assumed from Table XII. The pressure at the point *A*, the beginning of compression, depends on the type of engine but for the four-cycle in question may be assumed to be 13.2 absolute or 1.5 pounds below atmosphere. In two-cycle engines the pressure at *A* may be atmospheric or even 1.5 pounds above, depending on the pressure of the scavenging air.

It might be well before proceeding with the construction of the indicator card to discuss the clearance volume. At the point *A*, Fig. 146, the piston is at the outer end of the stroke and the cylinder is full of the mixture of gas and air in the case of a gas engine, or of pure air in the case of an oil engine. Compression takes

place from A to B along a curve whose equation is $PV^n = C$ where C is a constant and n has been found by experiment to have a value of 1.30 to 1.35 in the case of a gas engine and about 1.40 for an oil engine compressing pure air.

If we let V_a and V_b represent volumes and P_a and P_b pressures at the points A and B respectively, the compression ratio is

$$r = \frac{V_a}{V_b} = \left(\frac{P_b}{P_a}\right)^{\frac{1}{n}}$$

which may now be computed as P_a, P_b and n have all been assigned values to meet the conditions assumed. The per cent clearance is then

$$\frac{V_b}{V_a - V_b} = \frac{1}{\frac{V_a}{V_b} - 1} = \frac{1}{r - 1}$$

In the problem selected the compression ratio and clearance volume are

$$r = \left(\frac{P_b}{P_a}\right)^{\frac{1}{n}} = \left(\frac{154.7}{13.2}\right)^{\frac{1}{1.35}} = \overline{11.7}^{.741} = 6.20 \text{ and}$$

$$\text{clearance volume per cent} = \frac{1}{r - 1} = \frac{1}{6.2 - 1} = .1923 \text{ or } 19.23$$

per cent with the allowable compression pressure 140 lbs. gauge and $P_a = 13.2$ absolute. The values given in Table XII from which we selected compression ratio and clearance were worked out as above.

In order to show the values of compression ratios and per cent clearance volumes for various assumed values of P_b, P_a and r, the curves in Figs. 147 and 148 are given here.

Before the indicator card can be completed the pressure at x, the theoretical explosion pressure, must be determined as a starting point of the expansion line. The only known quantity with which we have to work to find the pressure at x is the mean effective pressure which the card must show on completion. The mean effective pressure is proportional to the area of the card divided by the length, and as the length, $V_a - V_b$, is fixed, the curve P_xP_d must be located so that the area between the two curves divided by the length of the card will be equal to the m.e.p. when multiplied by the proper scales. The area under xD is $\frac{P_xV_x - P_dV_d}{n - 1}$ and the area under the compression line is $\frac{P_aV_a - P_bV_b}{n - 1}$. Since the actual card will have rounded corners,

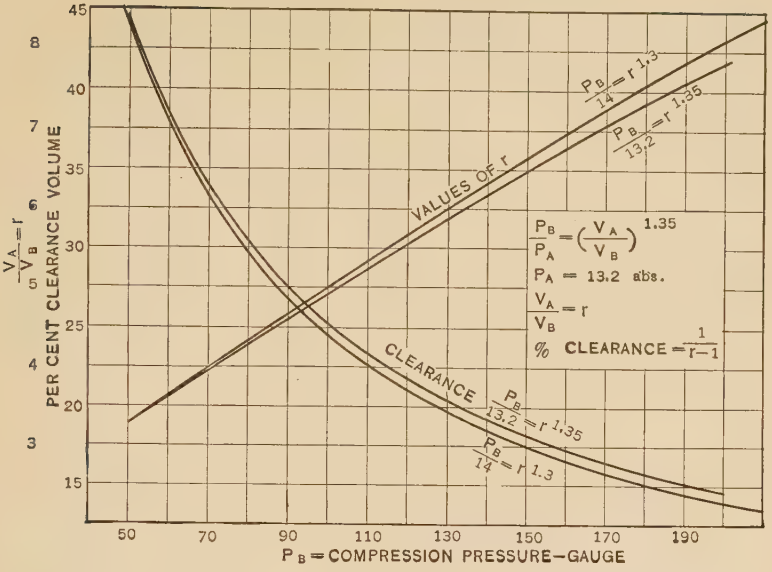


FIG. 147.

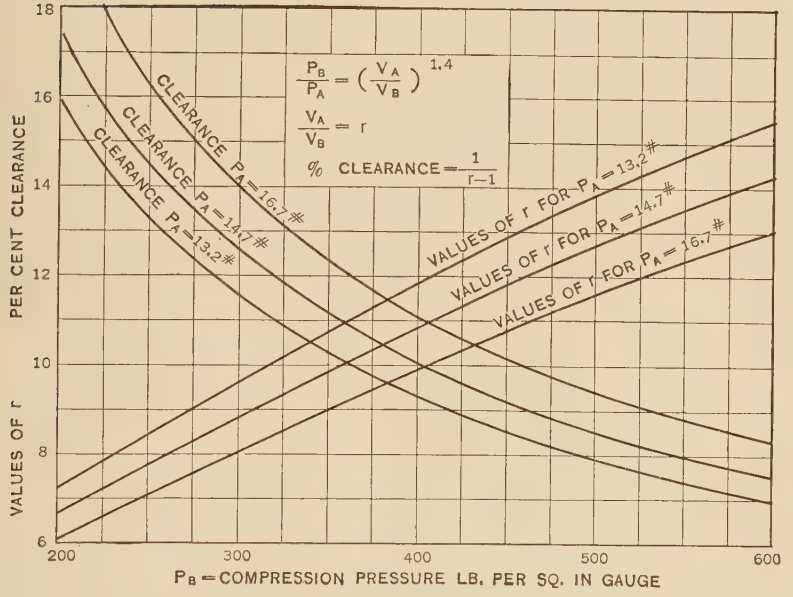


FIG. 148.

the actual area will be less than the theoretical card with square corners and we must make allowance for this difference by introducing a factor to take care of it. This factor, f, will be from 0.96 to 0.98. Then,

$$\frac{\text{m.e.p.}}{f} = \frac{\dfrac{P_xV_x - P_dV_d}{n-1} - \dfrac{P_vV_b - P_aV_a}{n-1}}{V_a - V_b}$$

where m.e.p. = pounds per square inch and P_a,P_x, etc., are pounds per square inch absolute.

Since combustion takes place at constant volume, $V_b = V_x$ and for convenience it may be assumed that $V_b = V_x$ = unity. Then $V_a = V_d = r$ and $V_a - V_b = r - 1$.

Hence

$$\frac{\text{m.e.p.}}{f} = \frac{\dfrac{P_xV_x - \dfrac{P_xV_x^n}{V_d^{n-1}}}{n-1} - \dfrac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b}.$$

This equation contains only one unknown P_x which may be solved for. The expansion line xD should be started from P_x and the compression line from P_b. The point of the card should be rounded off below P_x, say at 350 to 450 pounds, depending on the kind of fuel.

The expansion and compression curves may be plotted as follows:

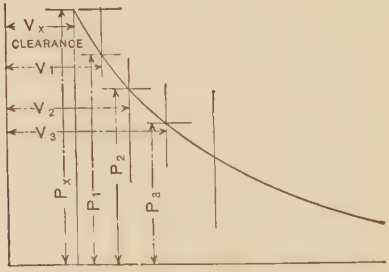


FIG. 149

$$P_xV_x^n = P_1V_1^n,$$

$$P_1 = P_x\left(\frac{V_x}{V_1}\right)^n. \qquad \text{Let} \quad \left(\frac{V_x}{V_1}\right)^n = K.$$

Let $\dfrac{V_x}{V_1}$ = constant = C. That is, select ordinates through 1, 2,

3, etc., so that the volume at each will be the volume at the preceding ordinate divided by constant C. Thus $V_1 = \frac{V_x}{C}$, $V_2 = \frac{V_1}{C}$, etc. Then the successive pressures may be calculated and plotted. $P_1 = KP_x$, $P_2 = KP_1$, etc., where $K = C^n$. By this method, illustrated in Fig. 149, it is necessary to raise only one number C to the nth power. The value of n may be assumed to be 1.35. The completed indicator cards should have the appear-

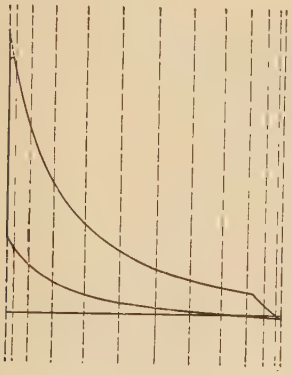


FIG. 150.

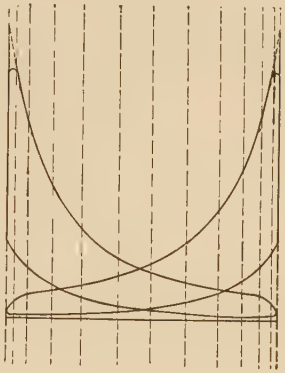


FIG. 151.

ance of those shown in Figs. 150 and 151, which represent cards from single- and double-acting engines respectively.

The Diesel Engine Indicator Card.—In constructing the indicator card for the Diesel engine, it will be necessary to determine the point of cut-off, or the point V_x, Fig. 152. Proceeding in a manner similar to that for the Otto card, we find the following equation to be true:

$$\frac{\text{m.e.p.}}{\text{diag. factor}} = \frac{P_b(V_x - V_b) + \frac{P_xV_x - P_dV_d}{n-1} - \frac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b}$$

$$= \frac{P_b(V_x - V_b) + \frac{P_bV_x - \frac{P_bV_x{}^n}{V_d{}^{n-1}}}{n-1} - \frac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b},$$

where m.e.p. is pounds per square inch as already found and P_a, P_b, etc., are pounds per square inch absolute.

Assume for convenience that

$$V_b = 1.$$

Then

$$V_a = V_d = r,$$

$$V_a - V_b = r - 1,$$

$$n = 1.35,$$

Diagram factor = 0.96 to 0.98.

Theoretically the value of n for the compression of pure air is not the same as the n for the expansion of the burned gases. Actually they are nearly the same and lie between 1.30 and 1.35, depending on the type of engine.

The value of $P_b = P_x$ is usually 500 pounds for a Diesel engine.

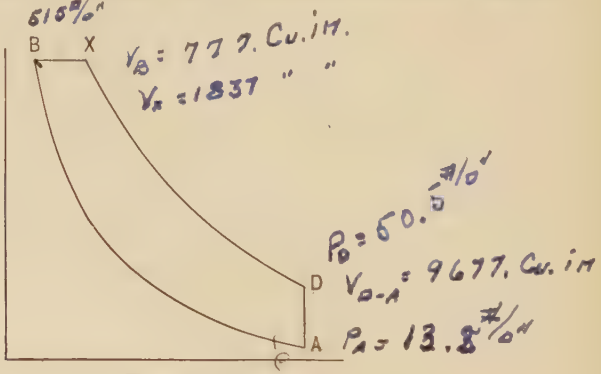


FIG. 152.

The pressure at the beginning of compression P_a will be atmospheric or 2 or 3 pounds above for a two-cycle engine, and will be 1.5 or 2 pounds below the atmosphere for a four-cycle engine. The value of the clearance should be found as for the Otto cycle engine.

As a matter of convenience, it will be well to make the indicator card about 4 inches long, and some even proportion of the stroke. Thus for a 16-inch stroke the card would be made one-fourth the stroke, and for a 24-inch stroke it would be made one sixth the stroke. The spring (vertical) scale used in drawing the indicator cards may be 100 pounds to the inch for all types of engines. The base-scale and the area-scale should be computed, the former being the number of feet on the stroke represented by each inch of length on the indicator card. The area-scale is the product of the spring- and base-scales and the unit is foot-pounds per square inch of indicator card.

The mean effective pressure of the finished card should be measured by a planimeter. The point of release must be assumed, however, before the card may be completed. The release should occur with the crank 30 to 40 degrees before dead center for a four-cycle engine. Release occurs on a two-cycle engine when the piston uncovers the exhaust-ports, the ports being made to give proper exhaust velocities. This will be taken up in a later chapter.

In a four-cycle engine release may be obtained on the card as shown in Fig. 153. The crank semi-circle should be laid off, as shown, the proper distance from the card. Draw in crank position as shown, 30 to 40 degrees before dead center and with a

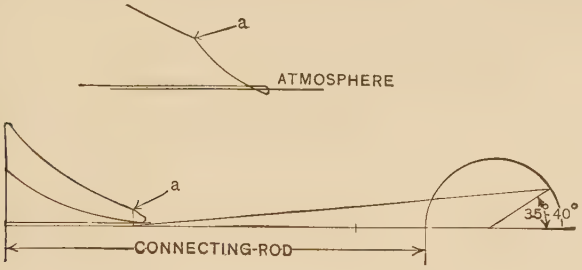


FIG. 153.

radius equal to the length of the connecting-rod, find the point of release on the stroke (card base) and project it up to the expansion line as at *a*, Fig. 153. From *a* to the end of the stroke the curve will be free expansion and will have the form shown in the large scale detail. It will, in some cases, go down below the atmosphere, particularly in engines with long exhaust pipes. In drawing the card with a spring scale of 100 pounds, it will be sufficient to carry this line down to the atmospheric line.

EXERCISES

1. Find the air required per cubic foot of gas of the following composition by volume:

	PER CENT
H_2	= 27.5
CO	= 45.0
O_2	= 3.0
CH_4	= 3.5
CO_2	= 7.0
N_2	= Bal.

2. Find the heating value of the gas given in (1) above.

3. Find (*a*) the heat available in one cubic foot of mixture of the above gas, normal temperature and pressure. (*b*) Find the ft.-lbs. work per cubic inch. (Assume air stand efficiency 54 per cent, relative efficiency 63 per cent and volumetric efficiency 75 per cent.)

4. Find the size of the cylinder to develop 125 maximum indicated horse-power, $\frac{L}{d} = 1.4$, r.p.m. = 180, engine to be 4-cycle, single-cylinder, single-acting.

5. Find the pressure, P_x, for the starting point of the expansion line of an indicator card for an m.e.p. of 90 lbs., $V_b = 1$, $V_a = 6.3$, $P_b = 130$ lbs. gauge, $P_a = 13.2$ lbs. abs. Diagram factor = 95 per cent, $n = 1.35$.

6. Find the cylinder dimensions of an 80 brake horse-power, single-acting, four-cycle, single-cylinder, 200 r.p.m. engine. Fuel composition: $C = 83.5$ per cent, $H_2 = 14.5$ per cent, excess air 50 per cent, air standard efficiency 63 per cent, relative efficiency, 60.0 per cent, volumetric efficiency 76 per cent and mechanical efficiency 82 per cent. $\frac{L}{d} = 1.25$.

7. Find the mean effective pressure and V_x for the above engine.

CHAPTER XIX

DESIGN OF THE GAS ENGINE (Continued)

Piston — Piston-rings — Wrist-pin — Piston-rod — Crosshead — Connecting-rod

The Trunk Piston. — In a single-acting engine a **trunk piston** is usually used and more often than not it is used without a crosshead; that is, the piston takes the vertical component S of the connecting-rod thrust instead of a crosshead taking it. The bearing area between the trunk piston and cylinder wall is based on the maximum vertical component which should be found graphically, by trial. This maximum occurs usually between 30 and 40 degrees crank angle.

FIG. 154.

In order to allow for rings in the piston the total bearing area should be considered to be the product of 0.8 of its length by 0.85 of its diameter. The bearing area is then 0.68 DL', where D is the diameter and L' the length of the piston. The bearing pressure should not exceed 30 pounds per square inch of this net area, and where a long piston may be used without increasing the weight of reciprocating parts to a prohibitive figure, the pressure should be 20 pounds. Thus for fast running engines the higher value should be used and for slower engines the lower. In no case should the length of piston be less than the diameter. In **stationary** engines it is better to have length of piston $1\frac{1}{4}$ to $1\frac{1}{2}$ times the diameter.

In Fig. 154 is shown the construction for finding maximum value of S. This should be done for several values of θ, the value of P being taken from the indicator card corresponding to the

value of θ. After S_{max} is determined, the length of the piston may be found. $L' = \dfrac{S \times \text{Area piston}}{0.68\,Dp}$, where S = vertical component pounds per square inch of piston, L' = length of piston in inches, D = diameter piston in inches, and p = allowable bearing pressure in pounds per square inch of projected area.

Strength of Piston. — The **thickness** of the **face** of the piston may be found by considering it a flat round plate supported at the edges. The dangerous section of such a plate is along a diameter. This section is under the action of two moments acting in opposite directions. The first is due to the fluid pressure $\dfrac{\pi D^2}{4 \times 2} p_{max} = \dfrac{\pi R^2}{2} p_{max}$ acting at an arm $\dfrac{4}{3}\dfrac{R}{\pi}$ so that the moment is $\frac{2}{3} R^3 p_{max}$. The second results from the reaction at the edge, which is $\dfrac{\pi R^2}{2} p_{max}$ acting at a lever of $\dfrac{2R}{\pi}$, giving the moment $R^3 p_{max}$.

The resulting moment is $R^3 p_{max} - \frac{2}{3} R^3 p_{max} = \frac{1}{3} R^3 p_{max}$. (1)

The section modulus of a flat plate of a thickness t and diameter D is

$$\frac{Dt^2}{6} = \frac{Rt^2}{3}. \tag{2}$$

Equating the resisting moment to the bending moment we have

$$K_b \frac{Rt^2}{3} = \frac{R^3 p_{max}}{3}, \tag{3}$$

$$t = R\sqrt{\frac{p_{max}}{K_b}}, \tag{4}$$

where K_b is the allowable stress in bending.

Bach writes this formula

$$t \cong r\sqrt{n\frac{p_{max}}{K_b}},$$

where $n = \frac{6}{5}$ for supported plates and $\frac{4}{5}$ for fixed plates and r is the **inner** radius. In formula (4) we have assumed R to be $\dfrac{D}{2}$, the outer radius, and have assumed the plate to be supported instead of fixed. These assumptions make (4) extremely safe so that a high stress may be used, say 8500 pounds per square inch.

To make the piston stronger still, the head is often ribbed

inside with four or six ribs of a thickness one-third to one-half the thickness of the face. These ribs are radial, and may extend back to the open end of the trunk or may stop at the piston-pin boss.

The maximum thickness of the piston **barrel** may be found by the formula

$$t_1 = \left(\frac{D}{50} + t_r + 0.2\right) \text{inches},$$

where D = diameter cylinder in inches,
t_r = thickness ring in inches,
t_1 = thickness barrel.

The decrease of t_1 toward the open end depends on the webbing and on the considerations of manufacture. The thickness at end may be $\frac{1}{2}\,t_1$ to $\frac{1}{3}\,t_1$. To allow for expansion due to high temperatures the piston barrel should be tapered from the last ring-groove to the closed end. The amount of this taper varies with the size, kind of service and with the material. It may be found correctly by experience only, but is usually 0.2 to 0.5 per cent of the diameter.

The first **ring-groove** should be at a distance from the end of the piston at least equal to the piston-face thickness, better 1.3 times the face thickness. The thickness of the bridge between the grooves should be at least equal to the width of the ring. The depth of the ring-groove should exceed the thickness of the ring by not more than 0.02 inch. If a greater clearance is allowed, it fills up with oil and the oil thickens, sticking the ring fast in the groove.

To obtain uniform distribution of thrust between piston and cylinder wall, the **wrist-** or **piston-pin** should be placed in the center of the part of piston carrying the load. If the combined width of rings be subtracted from the total piston length, half the remainder will be the proper distance for the center of pin from the open end.

Piston-rings. — These rings are usually plain cast-iron snap rings of small width, fitted in grooves cut around the circumference of the piston. Special spring rings and detachable seats for the rings have not proven successful, as nearly all these special constructions lose their efficiency after a short time by burning or sticking fast. In general the snap rings are made of **uniform thickness** as **eccentric** rings do not present any great

advantage. On the other hand they cost more to make and toward the joint the clearance at the bottom of the groove is increased to a dangerous degree.

In Fig. 155 is shown a piston ring sprung to a diameter D, the bore of the cylinder. The pressure per square inch between ring and cylinder wall is p pounds. If we assume the width of

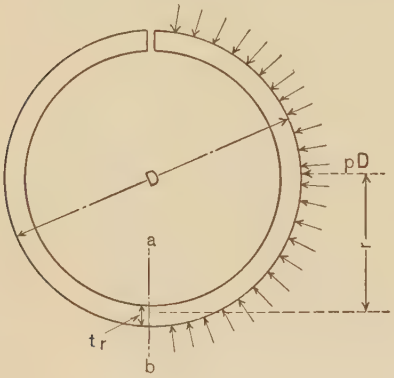


Fig. 155.

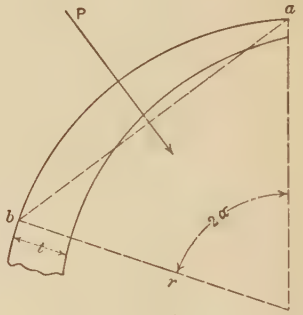


Fig. 156.

ring to be unity, then the pressure on half the circumference of the ring will be pD and this pressure will be exerted as indicated by pD in the figure. The moment of this force about the plane ab opposite the cut in ring is approximately $p\,Dr = 2\,pr^2$ where r is the radius of cylinder in inches.

The resisting moment of ring is $\frac{K_b t_r^2}{6}$, where t_r = thickness of ring in inches.

Then
$$2\,pr^2 = \frac{K_b t_r^2}{6}, \tag{1}$$

or
$$\frac{r}{t_r} = \sqrt{\frac{K_b}{12\,p}}. \tag{2}$$

From this equation the allowable thickness of the ring may be found. The allowable stress may be taken very high, 12,000 to 17,000 pounds per square inch for cast-iron.

Eccentric Rings. — In Fig. 156 is shown a short section of a ring of **varying thickness,** initially circular on the outside, which has been sprung into a cylinder of smaller diameter. Let I be the moment of inertia of the section at b, which may be taken

normal to the outside curve of the ring. Let ρ be the initial outside radius of the ring and t the thickness at b, all dimensions in inches. It will be necessary to make the ring of uniform width w. It is required to find the varying thickness t which will produce a uniform pressure of p pounds per square inch between the ring and cylinder walls.

The pressure of the ring on the cylinder per unit of length is pw, and the result of the uniform pressure from a to b is $pw \times$ chord ab or

$$P = 2\,pwr \sin \alpha. \tag{1}$$

The moment of this about b is

$$2\,pwr^2 \sin^2 \alpha. \tag{2}$$

Putting this value in the equation for the bending moment,

$$M = EI\left(\frac{1}{r} - \frac{1}{\rho}\right), \tag{3}$$

where $$I = \frac{wt^3}{12},$$

hence $$\frac{1}{r} - \frac{1}{\rho} = \frac{24\,pr^2 \sin^2 \alpha}{Et^3}. \tag{4}$$

For $2\,\alpha = 180$ degrees let t_r be the thickness of the ring.

Then $$\frac{1}{r} - \frac{1}{\rho} = \frac{24\,pr^2}{Et_r^3}. \tag{5}$$

Combining (4) and (5)

$$\frac{t}{t_r} = \sqrt[3]{\sin^2 \alpha}. \tag{6}$$

A table of values for t for different angles is given here:

$2\,\alpha$	t
10	$0.197\,t_r$
20	0.311
40	0.489
60	0.630
80	0.745
100	0.837
120	0.908
140	0.960
160	0.990
180	1.000

Fig. 157 shows a ring drawn from values given in the above table. About two-thirds of the inside curve agrees closely with a circular arc struck from the point C at a distance a from the center of the outside ring. It may be shown graphically that $a = 0.21\,t_r$,

approximately. The ring can be bored to a radius $r - 0.79\,t_r$ and the points near A thinned to agree with the proportions above.

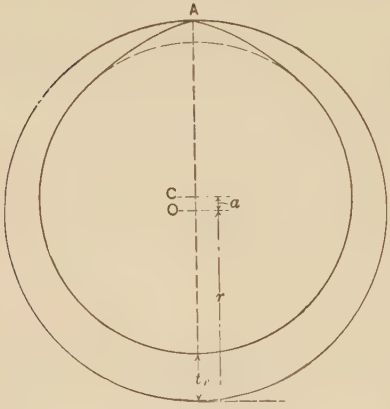


FIG. 157.

From equation (5)

$$p = \frac{Et_r^3}{24\,r^2}\left(\frac{1}{r} - \frac{1}{\rho}\right). \qquad (7)$$

The stress in the ring at its thickest part is

$$K = \frac{Md_1}{I} = \frac{6\,M}{bt_r^2}, \qquad (8)$$

$$M = EI\left(\frac{1}{r} - \frac{1}{\rho}\right) = \frac{Ebt_r^3}{12}\left(\frac{1}{r} - \frac{1}{\rho}\right).$$

Hence

$$K = \frac{Et_r}{2}\left(\frac{1}{r} - \frac{1}{\rho}\right). \qquad (9)$$

Using this value in (7)

$$K = 12\frac{pr^2}{t_r^2}, \qquad (10)$$

$$t_r = r\sqrt{\frac{12\,p}{K}} \qquad (11)$$

which is the same equation derived for rings of uniform thickness in the last article.

From equation (9) we get

$$\rho = \frac{rEt_r}{Et_r - 2\,Kr}, \qquad (12)$$

which gives us a value for the outside radius of the ring before it is sprung into the cylinder. The piece to be cut out is $2\,\pi\rho - 2\,\pi r$ and may be found from (12) as follows:

$$2\,\pi\rho - 2\,\pi r = \frac{4\,\pi Kr^2}{Et_r - 2\,Kr}. \qquad (13)$$

The stress due to slipping the ring over the piston to get it in the groove may be greater than that due to springing the ring into the cylinder. In slipping the ring over the end of the piston the outside radius of the ring is increased from ρ to R where $R = r + t_r$.

Then

$$\frac{1}{\rho} - \frac{1}{r + t_r} = \frac{2K'}{Et_r},$$

$$2K' = \frac{Et_r}{\rho} - \frac{Et_r}{r + t_r},$$

$$K' = \frac{Et_r(r + t_r - \rho)}{2\rho(r + t_r)}. \tag{14}$$

K' should not be greater than 17,000. In some cases it will be less than K, the stress due to springing the ring into the cylinder; in other cases it will be greater. Usually K' increases with increasing values of p, the unit pressure between the ring and the cylinder walls.

In order to show the application of the above formula, the rings for a 21-inch diameter cylinder will be calculated here. Assuming a pressure p of 4 pounds per square inch, a stress of 15,000 pounds per square inch, and $E = 17{,}000{,}000$ we find from (11)

$$t_r = 10.5\sqrt{\frac{12 \times 4}{15{,}000}} = 10.5 \times 0.657 = 0.59 \text{ inch.}$$

From (12) we find the value of

$$\rho = \frac{10.5 \times 17{,}000{,}000 \times 0.59}{17{,}000{,}000 \times 0.59 - 2 \times 15{,}000 \times 10.5} = 10.84 \text{ inches.}$$

The part cut out will be $2\pi(10.84 - 10.50) = 2.14$ inches.
The stress due to slipping the ring over the end of the piston may be found from (14),

$$K' = \frac{17{,}000{,}000 \times 0.59(10.5 + 0.59 - 10.84)}{2 \times 10.84(10.5 + 0.59)} = 10{,}430 \text{ pounds.}$$

If, instead of using 4 pounds per square inch of projected area between the piston rings and cylinder walls, we use 5 pounds, keeping the other values the same, we find the thickness of the ring = 0.66 inch, $\rho = 10.80$ inches, the part cut out = 1.88 inches and the stress $K' = 16{,}750$.

As stated before, the width of the ring has no influence on the stress, so that this dimension may be assumed arbitrarily. It may be assumed to be from $1\frac{1}{4}$ to 2 times the thickness t_r, depending on the number of rings used. The number of rings may be found by

$$n = \frac{D}{5w},$$

where D is the cylinder diameter and w the width of one ring, both in inches. It is of practical advantage to use narrow rings in sufficient number rather than wider rings, fewer in number.

Piston-pin. — The piston-pin should be designed to carry the maximum total load on the piston, that is, the load due to explosion pressure. The pin may be proportioned so the ratio of $\frac{l}{d} = \frac{4}{3}$, where l = length and d = diameter of pin. The **projected area** will then be ld. Assuming an allowable bearing pressure of 1700 to 2000, p', pounds per square inch of projected area, we have

$$p'ld = p_{\max}A, \tag{1}$$

where $p_{\max}$ = explosion pressure from card,
A = area cylinder in square inches.

If we use the ratio $\frac{l}{d} = \frac{4}{3}$, then

$$\tfrac{4}{3} \times p'd^2 = p_{\max}A, \tag{2}$$

which may be solved for d. Then $l = \frac{4}{3}d$, which should give a value $< \frac{D}{2}$ for convenient construction.

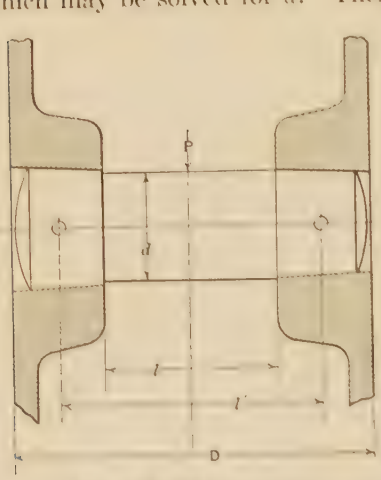


Fig. 158. — Piston-pin.

Strength of Pin. — The pin is fitted in the piston as shown in Fig. 158. We may assume it to be uniformly loaded as a beam for the distance l, with supports at the centers of the bosses at either end, making the length of beam between supports l'. Since l has already been determined, l' may be found from the relation

$$l' = l + \frac{D - l}{2} = \frac{l + D}{2}. \tag{3}$$

Since the load $p_{\max} \times$ area $= P$ is evenly distributed over the length l, the maximum bending moment will be at the center of the pin, or

$$M = \frac{P}{2}\left(\frac{l'}{2} - \frac{l}{4}\right). \tag{4}$$

This should be equated to the resisting moment of the pin in bending $K_t \frac{\pi d^3}{32}$, and the stress K_t solved for. This should fall below 12,000 pounds per square inch.

For the 21-inch cylinder of the specimen problem, assuming an explosion pressure of 425 pounds, we would have

$$\tfrac{4}{3} \times 2000\, d^2 = 425 \times 346,$$

$$d = \sqrt{\frac{3 \times 425 \times 346}{4 \times 2000}} = 7.43 \text{ or } 7\tfrac{7}{16} \text{ inches},$$

$$l = \tfrac{4}{3} \times 7.43 = 9.92 \text{ or } 10 \text{ inches},$$

$$l' = \frac{10 + 21}{2} = 15.5 \text{ inches},$$

$$M = \frac{425 \times 346}{2}\left(\frac{15.5}{2} - \frac{10}{4}\right) = 386{,}000 \text{ inch-pounds},$$

$$K_t = \frac{32 \times 386{,}000}{\pi \times 7.43^3} = 9410 \text{ pounds per square inch.}$$

The **piston-pin** may be held in place in the piston in any one of several ways. Some of the different ways are illustrated in Figs. 159 to 162. In Fig. 159 are shown two **set-screws,** with locknuts. This is an inexpensive and convenient method of holding the pin in small engines with a solid-end connecting-rod. Where the connecting-rod has a wedge adjustment, the set-screws interfere with the use of wrenches to take up such adjustment. The diameter of the set-screw may be $0.2\ d + \frac{3}{16}$ inch, where d = diameter of piston-pin.

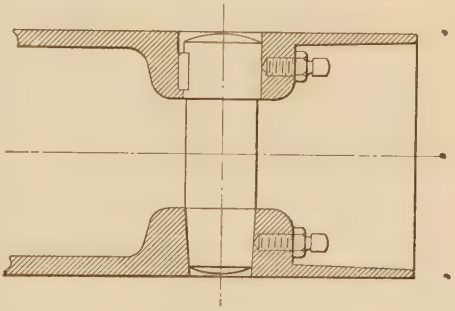


Fig. 159.

The **taper pin** shown in Fig. 160 may be used in preference to the set-screws when set-screws interfere with adjusting the rod. The diameter of the pin may be $0.15\ d + \frac{3}{16}$ inch. The taper pin and the set-screws are both unsatisfactory for fixing the piston-pin in large pistons. Much better ways are shown in Figs. 161 and 162. The method used in Fig. 161 is liable to spring the

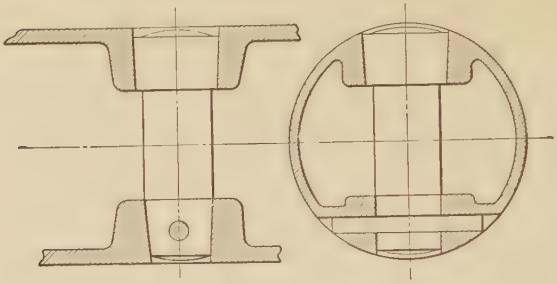


FIG. 160.

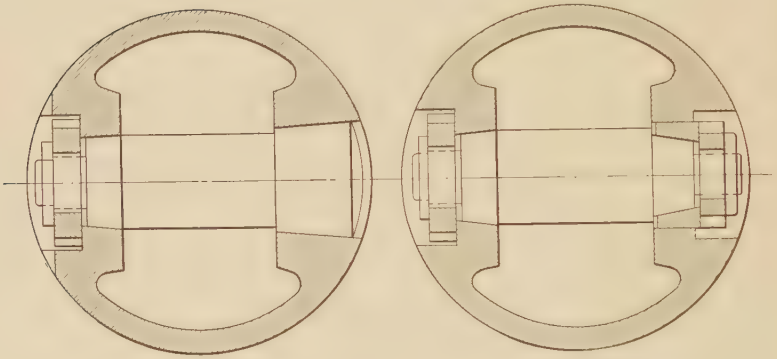


FIG. 161. FIG. 162.

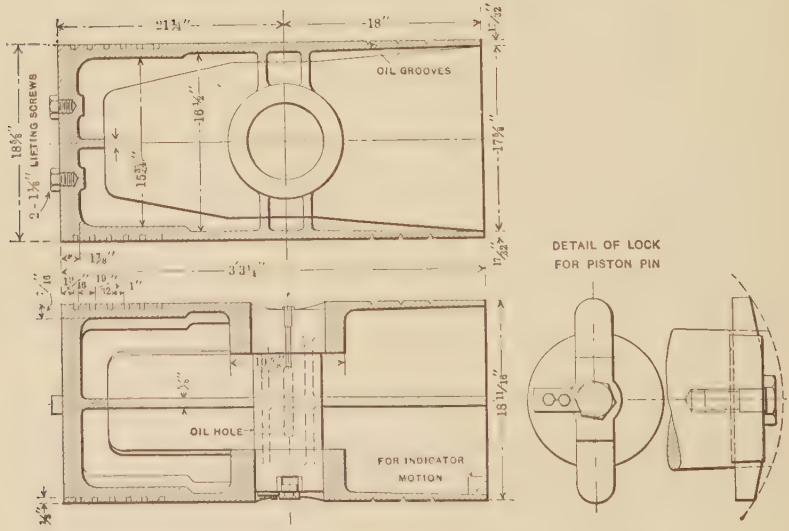


FIG. 163.—Typical Gas Engine Piston.

piston in large engines so that the method shown in Fig. 162 is often used.

A typical gas-engine **piston** is shown in Fig. 163 for a cylinder $18\frac{5}{8}$ inches diameter. The piston is ribbed and has oil-grooves near

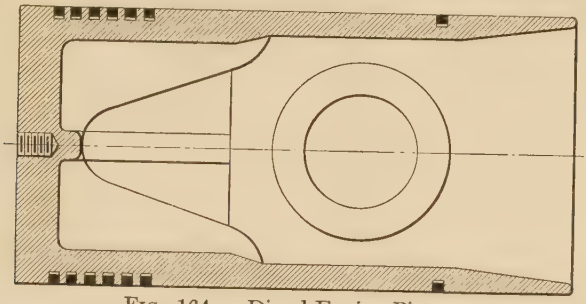


FIG. 164. — Diesel Engine Piston.

the open end instead of a snap ring used by many manufacturers. The pin is retained as shown in detail at the right of the figure. This method is not good for small engines.

The diameter of the boss may be made $1\frac{1}{2}\,d + 1$ inch, where d is the diameter of the pin.

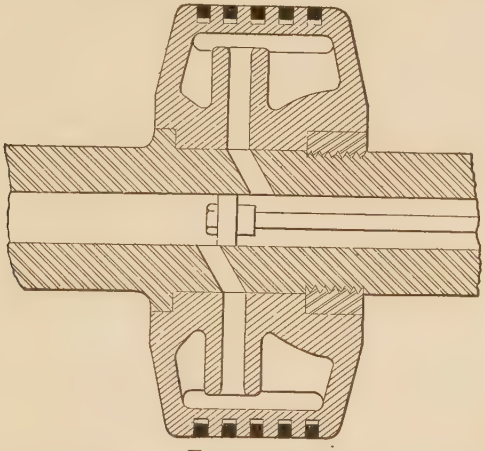


FIG. 165.

A **Diesel** engine piston is shown in Fig. 164. Here a cast-iron ring is used to retain the oil in place of the grooves shown in Fig. 163. This piston illustrates effectively the massive construction necessary in an engine of the Diesel type where pressures in starting are just about double the pressures found in a gas engine.

Double-acting pistons are illustrated in Figs. 165 and 166. This type of piston is always cooled with water as there is no chance whatever for radiation or conduction to the atmosphere direct. The water is conducted to the piston by means of the hollow piston-rod. Water enters the **bottom** and leaves at the top of the piston. This arrangement requires that there shall be a stop in the hole in the rod between the inlet and outlet openings at the piston. In Fig. 165 this stop is a separate piece held in place by a rod extending to the crosshead. In Fig. 166 the stop is formed by metal left at that point when the hole is drilled in the rod.

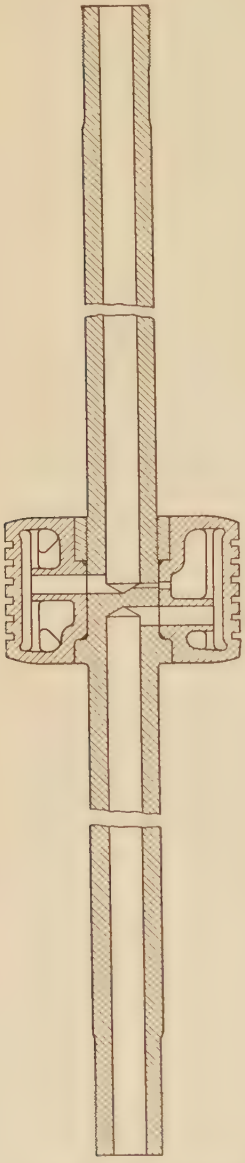


FIG. 166.

The **thickness** of the double-acting piston may be a little less than the thickness of the head of a trunk piston. Care must be taken to avoid straight lines and sharp corners. In other words, the casting must be as flexible as it is possible to make it. The piston is fixed to the rod by means of a nut which forces the piston against a shoulder on the rod. This nut is made round where it enters the piston, with a hexagonal head. After the nut has been tightened up as far as possible, the head is turned off flush with the piston. The collar on the rod may be designed for shear, but in such case a low fibre stress should be used. The diameter of the collar may be $1\frac{1}{4}\,d + \frac{1}{4}$ inch, where d is the rod diameter. The rod must be smaller on one end than the other in order to allow the threads to pass through the piston. The **threads** are usually made fine, say 8 to the inch, and of U. S. Standard form.

Piston-rod Design. — The **piston-rod** of a gas engine is subjected to alternate tension and compression and as the stresses alternate every stroke, the repetitions are frequent enough to make the application of the principles of fatigue a necessity. For this reason a low fibre stress should be used. The **load** on the rod is equal to the area of the piston multiplied by the explosion pressure.

$$\frac{\pi D^2}{4} \times p = K_t \frac{\pi d^2}{4}. \tag{1}$$

$$d = D\sqrt{\frac{p}{K_t}}. \tag{2}$$

In our specimen problem the diameter of the cylinder was found to be 21 inches, and explosion pressure of 425 pounds may be assumed for convenience. Then

$$d = 21\sqrt{\frac{425}{6000}} = 5\tfrac{3}{4} \text{ inches.}$$

This diameter is for a **solid rod.** To find the diameter of a **hollow rod** to correspond, it is only necessary to assume the diameter of the hole and add the area of the hole to the area of a $5\frac{3}{4}$-inch circle. The diameter corresponding to the sum of these areas will be the outside diameter of the hollow rod. The hole in the rod should be from 2 to 6 inches, depending on the size of the piston. In this case if we assume a 3-inch hole, the necessary outside diameter of the hollow rod will be $6\frac{1}{2}$ inches.

The stress in this rod, acting as a fixed-ended column, should also be found. Before this can be done the length of the rod must be assumed, and this assumption may be 2 × stroke. This would give us a rod 63 inches long for the engine under consideration. Using the formula

$$K = \frac{P}{A}\left(1 + \frac{1}{25{,}000}\,\frac{L^2}{\rho^2}\right),$$

where $\rho^2 = \dfrac{\text{moment of inertia of section of rod}}{\text{area of section}}$,

P = total load on rod = $p \times$ area piston,

A = cross-sectional area of rod,

L = length of rod in inches,

we find the stress to be 6150 pounds per square inch. This should not exceed 7000 pounds.

The **material** used in the piston-rods should be of a good grade open-hearth steel, with or without nickel. Some manufacturers use $3\frac{1}{2}$ per cent nickel steel; others use steel without nickel. In Germany it is the practice to machine the rods with a slight upward camber to compensate for the deflection due to weight of piston and rod. This is of doubtful value.

Crosshead.—All double-acting and many single-acting gas engines are built with **crossheads.** The office of this part of the engine mechanism is to provide a flexible joint between piston-rod and connecting-rod, also to take care of connecting-rod thrust. The amount of this thrust and the area of slide necessary to take care of it may be determined as in the trunk piston. In some cases of low-speed engines, slightly higher bearing pressures may be allowed on the slides than were allowed in the trunk pistons.

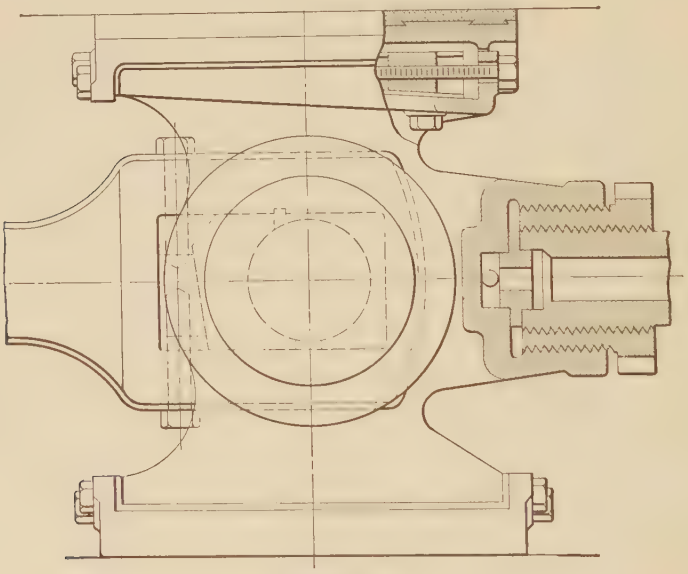


Fig. 167.

Types of **main crossheads** are shown in Figs. 167 and 168. The chief difference between the two is this: in the one shown in Fig. 167 the wear of the shoe is adjusted by means of the wedge shown; in Fig. 168 the wear is taken up by placing shims under the shoes. In both types the rod is secured in the cross-

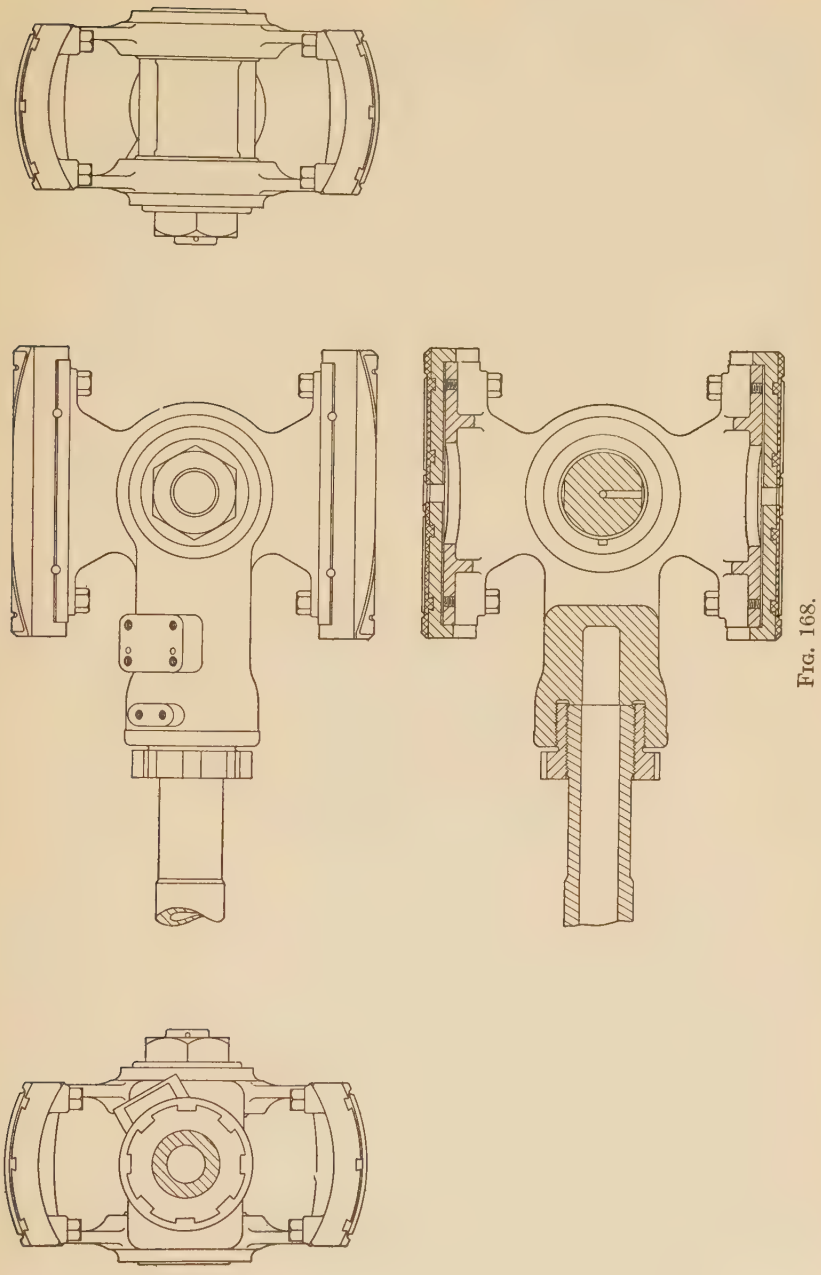
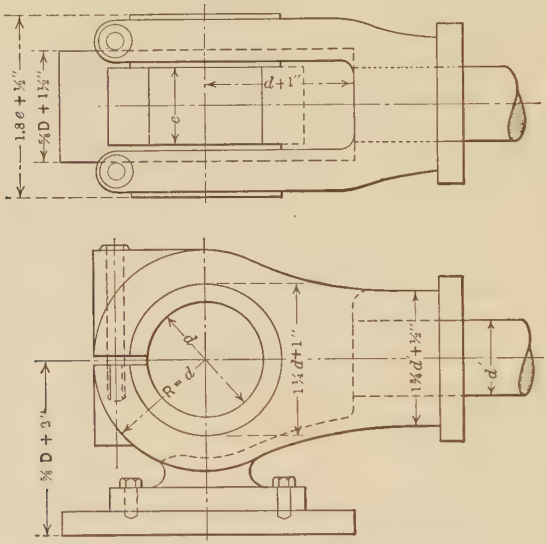


Fig. 168.

head by means of a nut threaded inside and outside. This nut is screwed into the crosshead as it is being screwed onto the rod. Since one of the threads is left hand it is obvious that the rod, crosshead and nut may all be locked together securely without employing any means to lock the nut in position.

Both the crossheads shown are of the **bored guide** type. The **marine** type, with one guide, is illustrated in Fig. 169. This type is lighter and consequently cheaper than the bored type



D = Dia. of Cylinder in inches.

Fig. 169.

but appears to lack the stiffness of the latter. The lower guide is the working surface with the engine running forward. This surface may be made ample to take any connecting-rod thrust. The upper surface takes the thrust of the rod when compression occurs in either end of the cylinder after the expansion pressure in the opposite end of the cylinder has decreased to a value below the compression pressure. The crosshead will then bear on the upper guide or slide. This slide is necessarily smaller than the bottom slide and the wear is often excessive. Premature ignition also puts a greater load on the upper slide than it should carry. For these reasons many manufacturers prefer the bored type of crosshead for double-acting engines.

The **stresses** in a crosshead are very complex and it is impossible to get a satisfactory solution, so that rational design gives way to empirical formula and data based on experience. The proportions given in Fig. 169 are applicable to many cases.

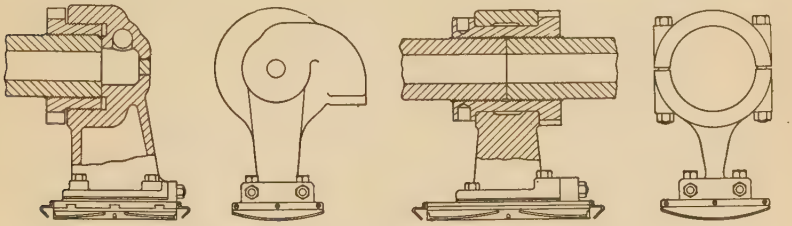


Fig. 170.

Crossheads of either type should be made of steel castings if possible, as they may be made lighter and more reliable than if made of cast-iron.

The **intermediate crosshead** of a tandem engine is illustrated

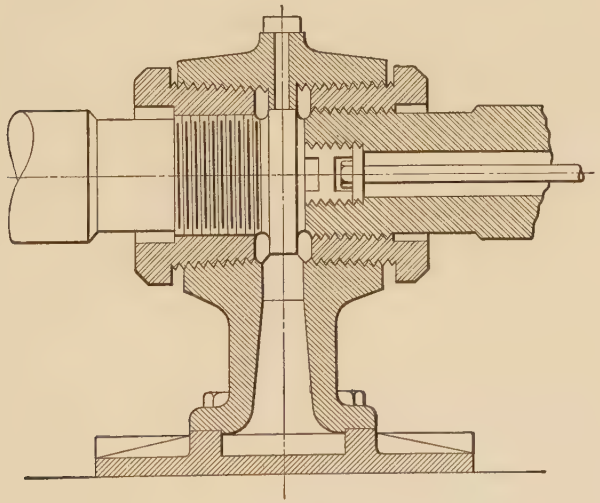


Fig. 171.

in Figs. 170 and 171. This crosshead is nothing more than a coupling for the piston-rods of the two cylinders with slide area enough to carry the weight of one piston and the rod reaching from piston to piston. The pressure per square inch on the slide

will be less than on the main crosshead. In the crosshead in Fig. 170 a single nut is used to pull the ends of the rod together. This nut is right and left hand. After the rods are thus joined, the split crosshead boss is attached by clamping. The crosshead illustrated in Fig. 171 has a solid boss and the ends of the rods are forced into this boss by a separate nut on each rod. The outside and inside threads of these nuts are right and left hand respectively, or vice versa.

This intermediate crosshead is used sometimes for admitting cooling water to the rods, in which case the water flows from this crosshead in both directions and is discharged at the main and tail crossheads. In other cases the water enters the main crosshead and passes through the pistons and rods in series, being discharged at the tail crosshead. The intermediate and **tail crossheads** in Fig. 170 are from an engine cooled in this way. The discharge passage on the tail crosshead is plainly shown. In the tail crosshead in Fig. 172, the water is discharged from a hole in the rod proper, shown ahead of the crosshead. A clamp is placed around the rod at this hole, the clamp having a passage to conduct the water to the waste passage.

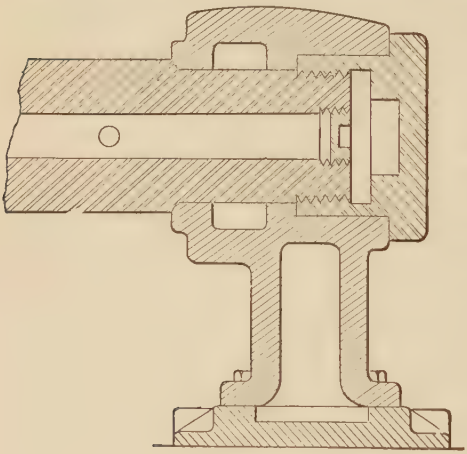


FIG. 172.

Both the intermediate and tail crossheads are simple castings of iron or steel. If of steel they may be made lighter than iron, but, as is true of most steel castings, they should be annealed. These parts are designed empirically and are made heavier than is necessary, usually. The thickness of metal should be ample for stiffness but care should be taken to keep the weight as small as possible. Adjustment is usually secured by means of shims.

The Connecting-rod. — Connecting-rods are made of steel forgings or castings. The wrist-pin and crank-pin ends should both be made adjustable so as to permit taking up the wear in

those bearings. In small engines the wrist- or piston-pin end is often made in the form of a solid eye with a bronze bushing. This is done because it is sometimes inconvenient to allow for adjustment on account of lack of space for bolts, keys, cotters, etc.

The design of the connecting-rod involves complicated calculations if exact results are to be expected. The chief complication arises when the stresses due to inertia of the rod are to be ascertained. The rod should be designed first, for direct compression at the small or piston end; second, as a pin-ended column in plane of rotation; third, for stress due to whipping action or bending due to inertia of rod.

It is quite common practice to make the connecting-rod rectangular in section, the larger dimension being that in the plane of rotation. In this case the rod should be investigated as a fixed-ended column in a plane normal to the plane of rotation.

The first step in getting the actual dimensions of the body of the rod is to determine the size at the small end for direct compression. The maximum axial force along the rod in a gas-engine is with the piston on the dead center. This force will be the area piston multiplied by the explosion pressure, when the engine first starts. After the engine gets up to speed this force will be diminished by the force of the inertia of the reciprocating parts. This force of inertia should not be taken into account, however.

The area of the small end will be the total force P divided by the allowable fibre stress for steel in alternate tension and compression. About 6000 pounds is a good value to use. If the rod is to be round, the following equation may be used to give the diameter:

$$\frac{P}{K} = \frac{\pi d^2}{4}, \tag{1}$$

$$d = \sqrt{\frac{4P}{\pi K}}. \tag{2}$$

If the rod is to be rectangular in section, the depth should be from $1\frac{1}{2}$ to 1.8 times the width. If we assume b, the depth, to be 1.75 times a, the width of the section, then

$$\frac{P}{K} = ab = 1.75\,a^2, \tag{3}$$

$$a = \sqrt{\frac{P}{1.75\,K}}. \tag{4}$$

In designing the rod as a **column** in the plane of rotation, we may use the Gordon formula

$$K = \frac{P}{A}\left(1 + C\frac{L^2}{\rho^2}\right), \tag{5}$$

where P is the load on the column, A the sectional area, L the length in inches, ρ the radius of gyration, and C is equal to $\frac{4}{25{,}000}$ for a pin-ended column. If the width of the section a be kept constant throughout the length of the rod, the other dimension b' will be the only unknown and will occur in the first power in A and in the second power in ρ^2. This dimension b' will be the depth of the rod halfway between the wrist-pin and crank-pin. The rough dimensions of the rod will be as shown in Fig. 173. L,

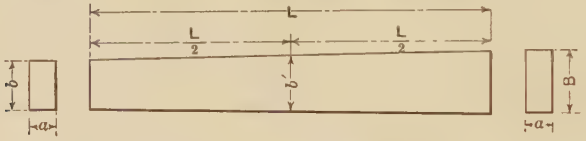


Fig. 173. — Connecting-rod Body.

the length, will be 5, $5\frac{1}{2}$ or 6 times the crank radius, depending on the ratio of $\frac{L}{R}$ chosen for the engine. The depth at the center of the crank-pin will be

$$B = b + 2\,(b' - b) = 2\,b' - b. \tag{6}$$

After the stub ends have been designed the dimensions b and B ends will be modified to suit the end construction, but the dimensions of the rod body will not change. The stress in a rod of uniform cross-section due to whipping, or, in other words, due to the inertia of the rod in the plane of rotation, may be found by the formula derived by Bach,

$$K_b = 0.000002\,N^2 r A w \frac{L^2}{Z}, \tag{7}$$

where

N = revolutions per minute,
r = radius of crank in inches,
A = area mean section of rod in square inches,
w = weight of rod material in pounds per cubic inch,
L = length of rod in inches,
Z = section modulus of rod.

For a rectangular rod $Z = \frac{bh^2}{6}$, then for a rod of that shape

$$K_b = 0.000012\, N^2 r w \frac{L^2}{b'}, \tag{8}$$

where b' is the depth of rod at the middle section.

In order to get the total stress in the rod the stress as found above should be added to that for direct tension or compression.

$$K_{\text{total}} = K_c + K_b = \frac{P}{ab'} + K_b. \tag{9}$$

The stress in the rod as a column at right angles to the plane of rotation will be

$$K' = \frac{P}{A}\left(1 + \frac{1}{25{,}000}\frac{L^2}{\rho'^2}\right), \tag{10}$$

where

$$\rho'^2 = \frac{\frac{a^3 b'}{12}}{ab'} = \frac{a^2}{12}.$$

Example. — To illustrate the design of the body of the rod we will assume a cylinder diameter of 21 inches, a stroke of 30 inches, an explosion pressure of 425 pounds per square inch and 150 r.p.m. The total force P will be 147,000. If we assume

$$b = 1.7\, a, \quad \text{then} \quad \frac{147{,}000}{6000} = 1.7\, a^2,$$

$$a = \sqrt{\frac{147{,}000}{6000 \times 1.7}} = 3.82 = 3\tfrac{13}{16} \text{ inches},$$

$$b = 1.7 \times 3.82 = 6\tfrac{1}{2} \text{ inches}.$$

To find b', by (5)

$$6000 = \frac{147{,}000}{3.82 \times b'}\left(1 + \frac{4}{25{,}000} \times \frac{90 \times 90 \times 12}{b'^2}\right),$$

$$0.1555\, b' = 1 + \frac{15.43}{b'^2},$$

$$0.1555\, b'^3 - b'^2 = 15.43.$$

Solving this equation for b' we find the depth of the rod at the middle to be 8 inches. By proportion we find the depth at the crank-pin to be 9 inches.

The stress due to whipping will be approximately

$$K_b = 0.000012 \times 150 \times 150 \times 15 \times 0.282 \times \frac{90 \times 90}{8}$$

$$= 1160 \text{ pounds per square inch.}$$

$$K_{\text{total}} = \frac{147{,}000}{8 \times 3.82} + 1160 = 5960 \text{ pounds per square inch.}$$

The maximum stress due to whipping does not actually occur at the center of the rod but at a place nearer the crank-pin, where the rod is of larger section. But as this location is difficult to determine before the actual shape and weight of the rod is known, we assume it to be at the middle of the rod, which is on the side of safety.

The stress in the rod assuming the rod to be a fixed-ended column in a plane normal to the plane of rotation will be

$$K' = \frac{147{,}000}{3.82 \times 8}\left(1 + \frac{1}{25{,}000} \times \frac{90 \times 90 \times 12}{3.82 \times 3.82}\right)$$

$$= 6080 \text{ pounds per square inch.}$$

Stub Ends. — A connecting-rod for a large double-acting gas engine is illustrated in Fig. **174.** The rod and straps are forged

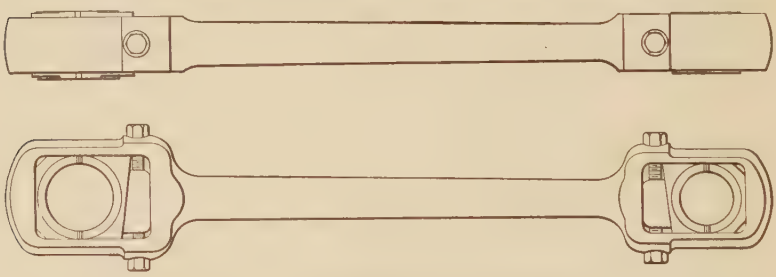


Fig. 174. — Forged-steel Connecting Rod.

in one piece. Babbitted shells or boxes are used at both ends, adjustment being secured by means of wedges. In this case adjustment at either end tends to lengthen the rod. The rod shown in Fig. 175 is a steel casting, with straps cast solid with the rod. The babbitted shells are wedge-adjusted in this case also, but adjustment tends to lengthen the rod at the wrist-pin end and shorten it at the crank-pin end. One noticeable feature of this rod is the I section which is difficult to secure in a forged rod but not difficult in a cast rod. Automobile rods which are I-shaped are usually drop-forged.

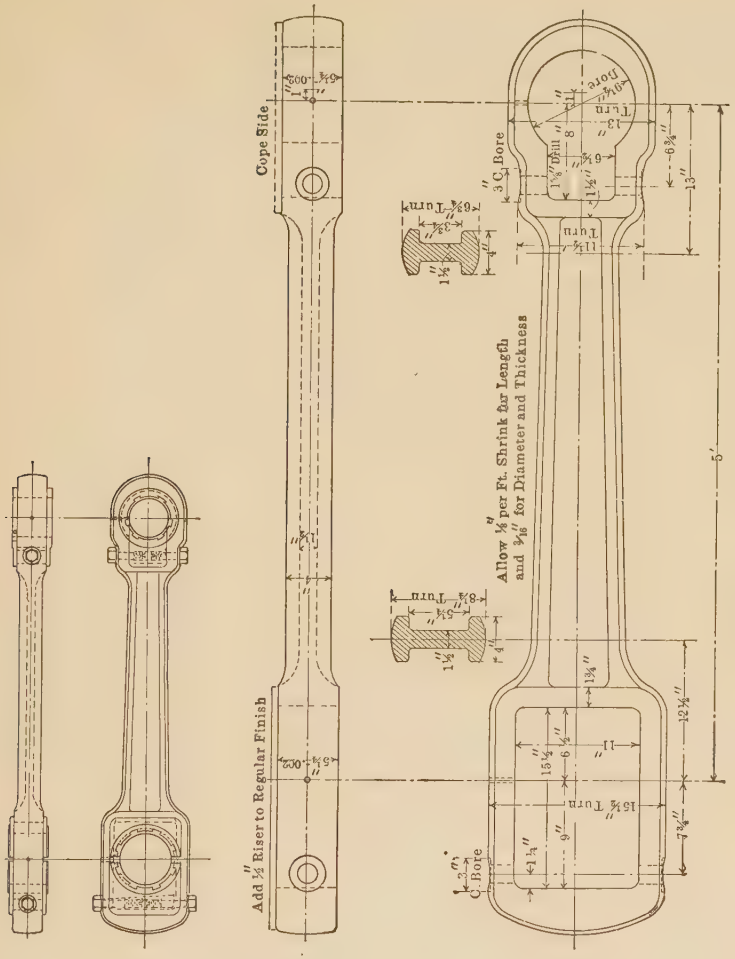


Fig. 175. — Cast-steel Connecting-rod.

The diameter and length of the wrist-pin have already been computed. These two dimensions form a basis for the calculations that follow. The width of the bearing is obviously equal to the effective length of the pin. The width of the straps that enclose the bearing may be from $\frac{1}{2}$ to $1\frac{1}{2}$ inches less than the effective length of the pin. Thus in Fig. 176, c may be made from $e - \frac{1}{2}$ inch to $e - 1\frac{1}{2}$ inches. The thickness of the strap a may be found by computing each side to take half the total piston pres-

sure P. Thus $\frac{P}{2K} = a\,(c - \delta)$. In this case a stress of 7000 pounds might be used for K.

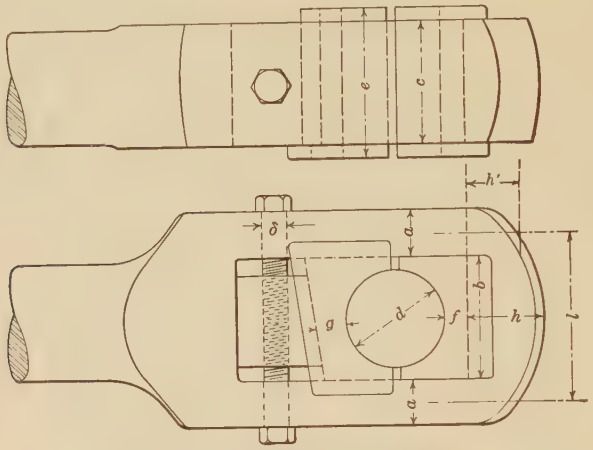


Fig. 176. — Wrist-pin Stub End.

To find the thickness h at the end of the rod we must first find b. The value b may be assumed to be $d + (\frac{1}{2}$ to $1\frac{1}{2}$ inches), the constant depending on the size of the engine. The end of the rod is similar to a beam with a uniform load whose total value is P. The bending moment is then $\frac{Pl}{8}$, where $l = b + a$. The value h will be found from the equation

$$\frac{Pl}{8} = \frac{Kch^2}{6}.$$

The stress may run up to 10,000 pounds in this part of the rod, if the above equation be used. The values of f and g vary between rather wide limits. Where bronze boxes are used these dimensions may be less than where babbitted shells are used. We might set the limits at $\frac{1}{2}$ inch and $1\frac{1}{2}$ inches, without being far wrong. The taper of the wedge should be one in six, and the diameter of adjusting screws will fall between $\frac{3}{4}$ inch and $1\frac{1}{2}$ inches.

The design of the crank-pin closed-strap end is not different from that given above. If a marine-end be employed, the design changes slightly. In Fig. 177 is shown a detail of the marine end such as is usually employed when a shaft with a center-crank

is used. Before the design can be carried to completion it will be necessary to find the approximate size of the crank-pin. Fig. 178

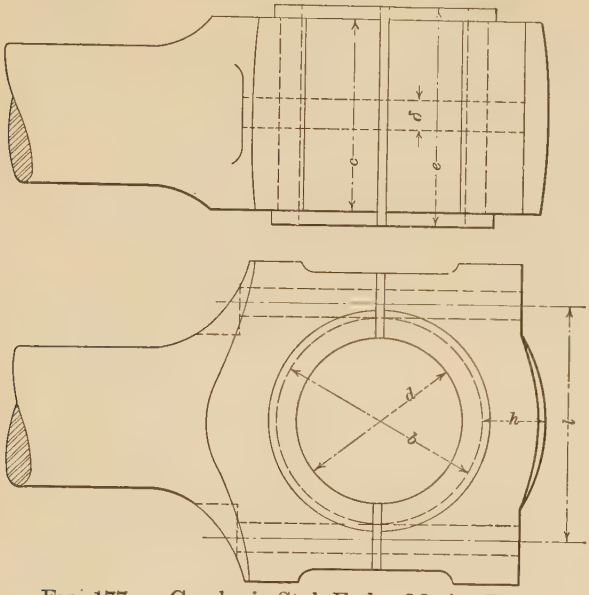


FIG. 177. — Crank-pin Stub End — Marine Type.

illustrates this step. The distance between the centers of the bearings may be assumed to be from 1.8 D to 2.2 D, where D is

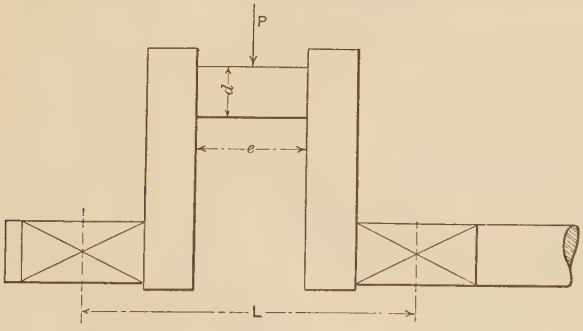


FIG. 178.

the cylinder diameter in inches. The diameter of the pin may be found from

$$\frac{PL}{4} = \frac{K\pi d^3}{32},$$

whence $$d = \sqrt[3]{\frac{32\,PL}{4\,K\pi}},$$

where $K = 12{,}000$ pounds per square inch. The length e may be made the same as the diameter. The bearing pressure $\frac{P}{d \times e}$ should be found and should fall between the values 1200 and 1500 pounds per square inch.

The design of a side crank-pin is a little different from that of a center crank-pin. It is convenient to assume the ratio of $\frac{e}{d}$, Fig. 179, in starting the design of a side crank-pin. Then the bending moment on the pin may be taken as $\frac{3}{4}\,Pe = \frac{3}{4}\,Pkd$, where $\frac{e}{d} = k$. Theoretically the pressure is evenly distributed over the length of the pin, giving a bending moment of $\frac{1}{2}\,Pe$. Actually the pressure may not be evenly distributed. Also, it would be too severe to say that the pressure is exerted at the end of the pin. Hence we use the mean value $\frac{3}{4}\,Pe$.

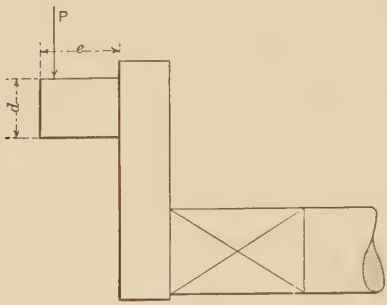


FIG. 179.

Then $$\frac{3}{4}\,Pkd = \frac{K\pi d^3}{32},$$

$$d = \sqrt{\frac{3\,P \times k \times 32}{4\,\pi K}}.$$

Here again the bearing pressure should be investigated and should fall between 1200 and 1500.

After the size of the pin has been determined, the other dimensions shown in Fig. 173 may be found. Thus, $b = d +$ ($\frac{1}{2}$ to $1\frac{1}{2}$ inches), and $l = b + \delta$. To get δ, assume each bolt to take its share of the total pressure on the piston.

$$\frac{P}{2} = \frac{K\pi d'^2}{4}, \qquad d' = \sqrt{\frac{4\,P}{2\,\pi K}},$$

where d' = diameter of bolt at root of thread,
$K = 12{,}000$.

The diameter of bolt δ will be d' + twice the depth of the threads. Fine threads are usually used on these bolts and the body of the bolt is turned down to the diameter of the root of the thread as shown in Fig. 180. This is done to prevent local strains. The part of the bolt that bears in the rod and cap should be a tight fit to line the parts up properly. These bolts are usually ground to a fit.

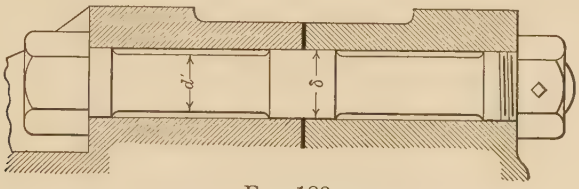


Fig. 180.

The nuts should be locked securely. The one in Fig. 180 is locked with a small set-screw tapped in the side. On smaller bolts double nuts may be used. Lock-plates are sometimes employed on large nuts.

It is best to get the bolts as close together as possible and in order to do this the bearing shell is cut away. In some cases the bolts are brought in until they just clear the pin. The thickness of the shell including babbitt should be from $\frac{3}{4}$ to $1\frac{1}{2}$ inches, depending on the size, when the shell is of cast steel. The thickness of the cap h may be computed from

$$\frac{P}{2}\left(\frac{l}{2}-\frac{b}{4}\right)=\frac{Kch^2}{6},$$

where c is from $\frac{1}{2}$ to $1\frac{1}{2}$ inches less than e.

The calculations above apply only to double-acting engines. The force that acts on the straps or crank-pin bolts of the connecting-rod for a single-acting engine is the inertia of the reciprocating parts. This force is so much less than the pressure due to the explosion that parts designed with it as a load would be very light compared to parts designed for the explosion pressure.

Fig. 181 illustrates a connecting-rod for a single-acting Diesel engine. The crank-pin cap is noticeably light, as are the bolts, compared to a double-acting engine. Another peculiarity is the wedge adjustment. On account of lack of room in the piston, the wedge is put in at right angles to the plane of the rod. This gives more room for using a wrench but shortens up the wedge.

Still another type of adjustment is shown in Fig. 182 which is the rod used in the Foos vertical multi-cylinder engine. This arrangement brings the adjustment outside the piston but costs

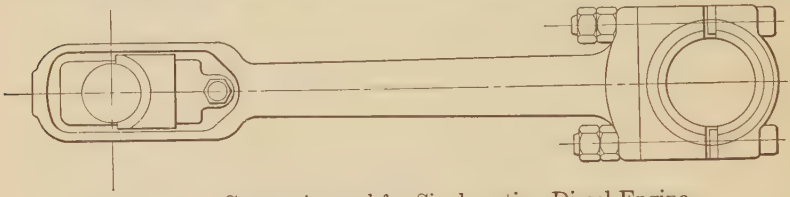


Fig. 181. — Connecting-rod for Single-acting Diesel Engine.

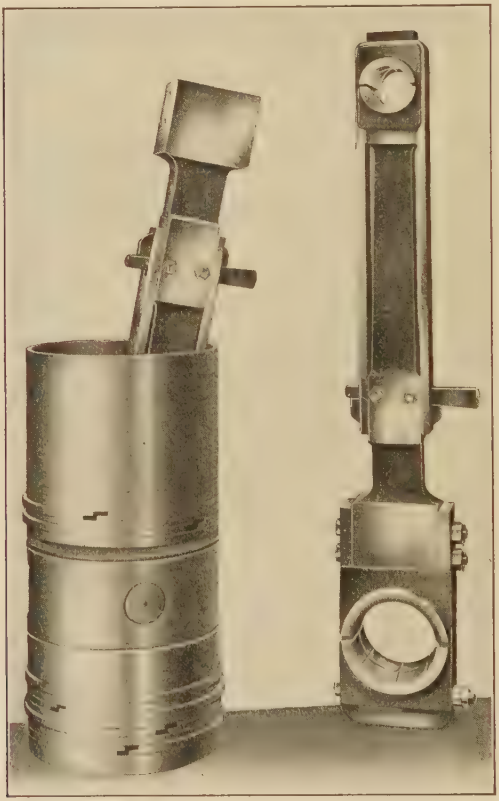


Fig. 182. — Foos Connecting-rod and Piston.

more to make than the one in Fig. 181. The crank-pin end shows the open-strap-end adapted to the center-crank. Both these adjustments tend to shorten the rod.

Some engineers do not use the method given on pages 304 and 307 for finding the thickness h of the connecting-rod end. Instead they design the end for shear, thus:

$$h' = \frac{P}{2\,K_s c},$$

where h' is the minimum thickness of the end of the rod, Fig. 176, P is the total maximum piston pressure in pounds, c is the width of the rod in inches and K_s is the allowable shearing stress, about 4000 pounds per square inch.

The thickness h at the middle is made from $\frac{1}{2}$ to 1 inch greater than h', depending on the size of the rod.

EXERCISES

1. Find the thickness of the head of a trunk piston 18 ins. diameter, pressure 425 lbs per sq. in. Material cast-iron.

2. Find the stress in the head of a trunk piston 20 ins. diameter, $1\frac{1}{4}$ ins. thick, 400 lbs. press.

3. Find the thickness of cast-iron piston rings for a 22-in. diameter cylinder, pressure between ring and cylinder 4 lbs. per sq. in. Find stress due to slipping ring into groove.

4. Find thickness of cast-iron piston rings for 22-in. diameter cylinder, pressure between cylinder and wall 5 lbs. per sq. in. Find stress due to slipping ring into groove.

5. Find the amount cut out of the rings in (3) and (4).

6. Design the piston-pin for the piston in (1), using bearing pressure and stress. Assume effective length of pin to be one-half cylinder diameter.

7. Design the piston-pin for a Diesel engine which has a cylinder 18 ins. diameter. Pin to be 9 ins. long.

8. Design the piston-rod for cylinder 20 ins. diameter, assuming length to be $2\frac{1}{4} \times$ diameter of cylinder. Section of rod to be solid circle. Maximum piston pressure 400 lbs. per sq. in.

9. Design piston-rod for a cylinder 18 ins. diameter, length of rod to be $2\frac{3}{4} \times$ diameter of cylinder. Rod to have hole whose diameter is one-half outside diameter of rod. Maximum piston pressure 400 lbs. per sq. in.

10. Find size of crosshead shoe for engine with cylinder 22 ins. diameter, maximum piston pressure 400 lbs. per sq. in. Vertical component of connecting-rod thrust to be $\frac{1}{10}$ total pressure on piston. Length of shoe $2 \times$ width.

11. Design rectangular connecting-rod for cylinder in Prob. 9, length of rod to be $4 \times$ diameter cylinder. Depth of rectangular section at the small end to be $1\frac{3}{4} \times$ the constant width. 160 r.p.m.

12. Find stress in rod in Prob. 11 in plane at right angles to plane of oscillation. Find stress due to whipping, if r.p.m. = 175, stroke = 24 ins.

13. Find the diameter of connecting-rod bolts, crank-pin end, of double-acting engine with 17-in. diameter cylinder, 425 lbs. maximum pressure.

14. Design crank-pin of center-crank type for engine in Prob. 13. Distance between center lines of bearings to be $2.1 \times$ diameter cylinder.

15. Design crank-pin of side-crank type for engine in Prob. 13.

CHAPTER XX

INERTIA OF RECIPROCATING PARTS — NET EFFORT DIAGRAMS — ROTATIVE EFFORT DIAGRAMS — WEIGHT OF FLYWHEEL — CONSTRUCTION OF FLYWHEEL — VELOCITY AND DISPLACEMENT DIAGRAMS

Weight of Reciprocating Parts. — In using the weight of reciprocating parts to find the effect of their inertia on the crank-pin pressure, the weight of the complete connecting-rod is usually included as a reciprocating part, though theoretically about half the weight of the rod has a rotating and not a reciprocating motion. In a tandem double-acting engine the weight of reciprocating parts will include the pistons, piston-rods, connecting-rod, main crosshead, intermediate crosshead and coupling, tail crosshead, water connections and cooling water in pistons and rods. The weights of the various parts may be computed from sketches made for each part as it is designed. For this computation the weight of cast-iron may be taken as 0.26 and steel 0.282 pound per cubic inch. Bronze weighs about 0.32 and brass 0.30 pound per cubic inch.

For approximate computations it is sometimes an advantage to be able to assume a weight of reciprocating parts. These may be assumed for the various engine types as shown in Table XIII.

Inertia Due to Reciprocating Parts. — The reciprocating parts of an engine have a velocity of zero at the ends of the stroke. In order to accelerate these parts up to their maximum velocity, which occurs somewhere near mid-stroke, the force of acceleration must be applied. The force of acceleration is supplied by the gas pressure, and reduces the effective pressure of the gas at the beginning of the stroke and increases it at the end of the stroke. In order to find an expression for the force of acceleration it is necessary to find first the expression for velocity. In Fig. 183 is shown the graphical construction which gives v, the piston velocity for the crank angle θ, when V is the constant velocity of the crank-pin.

TABLE XIII

WEIGHT OF RECIPROCATING PARTS PER SQUARE INCH OF PISTON

Kind of engine.	Weight per square inch of piston.
Single-acting:	
Trunk piston $L < 1.75\ D$	3.8 –4.2
Trunk piston $L > 1.75\ D$	4.2 –5.0
With crosshead	5.8 –7.2
Small high-speed automobile engines	0.45–0.75
Double-acting four-cycle:	
Single-cylinder	11–18
Tandem	18–25
Double-acting two-cycle	16–21

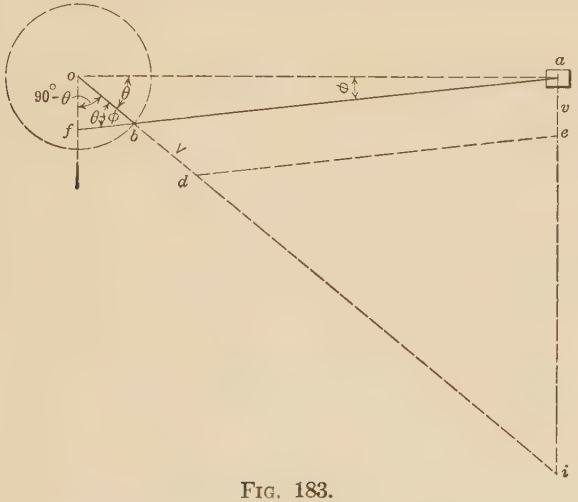


FIG. 183.

By similar triangles it may be shown that if the crank radius *ob* is assumed as the constant velocity of the crank-pin, the intercept on the vertical through *o* of the connecting-rod or the connecting-rod produced will give the velocity of the piston at that instant. Thus, in Fig. 183, *of* is the velocity of the reciprocating parts for the crank angle θ, when *ob* is the velocity of the crank-pin.

$$\frac{of}{ob} = \frac{\sin obf}{\sin ofb} = \frac{\sin(\theta + \phi)}{\cos \phi}.$$

For the outstroke

$$\frac{v}{V} = \frac{\sin(\theta + \phi)}{\cos \phi}.$$

For the return stroke

$$\frac{v'}{V} = \frac{\sin(\theta' - \phi')}{\cos \phi'}.$$

Then

$$\left.\begin{aligned} v &= V(\sin\theta + \cos\theta \tan\phi), \\ v' &= V(\sin\theta' - \cos\theta' \tan\phi'). \end{aligned}\right\} \quad (1)$$

When ϕ is a small angle, as is always the case in practical problems,

$$\tan\phi = \sin\phi = \frac{\sin\theta}{n}, \text{ approximately,}$$

where $n = \frac{L}{r}$ = length of connecting-rod divided by length of crank. Hence

$$\left.\begin{aligned} v &= V\left(\sin\theta + \frac{\sin 2\theta}{2n}\right), \\ v' &= V\left(\sin\theta' - \frac{\sin 2\theta'}{2n}\right). \end{aligned}\right\} \quad (2)$$

If α = acceleration in feet per second per second, then

$$\alpha = \frac{dv}{dt} = \frac{dv}{d\theta}\frac{d\theta}{dt} = \omega\frac{dv}{d\theta} = \frac{V}{r}\frac{dv}{d\theta}.$$

Then, using equation (2),

$$\left.\begin{aligned} \alpha &= \frac{V^2}{r}\frac{d}{d\theta}\left(\sin\theta + \frac{\sin 2\theta}{2n}\right) = \frac{V^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{n}\right), \\ &\frac{V^2}{r}\frac{d}{d\theta'}\left(\sin\theta' - \frac{\sin 2\theta'}{2n}\right) = \frac{V^2}{r}\left(\cos\theta' - \frac{\cos 2\theta'}{n}\right). \end{aligned}\right\} \quad (3)$$

The force of acceleration is the product of the mass into the acceleration, or $F = M\alpha$. For the sake of convenience we use the weight of reciprocating parts per square inch of piston area. Then

$$\left.\begin{aligned} F &= \frac{W}{Ag}\frac{V^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{n}\right), \\ F' &= \frac{W}{Ag}\frac{V^2}{r}\left(\cos\theta' - \frac{\cos 2\theta'}{n}\right). \end{aligned}\right\} \quad (4)$$

where W = total weight of reciprocating parts in pounds,
A = piston area in square inches,
V = crank-pin velocity in feet per second,
r = crank radius in feet,
g = 32.2.

TABLE XIV

Values of Terms in Expression for Force of Inertia

θ degs.	Cos θ	Cos 2 θ	$\frac{L}{r} = 4$		$\frac{L}{r} = 4\frac{1}{2}$		$\frac{L}{r} = 5$		$\frac{L}{r} = 5\frac{1}{2}$		$\frac{L}{r} = 6$	
			$\frac{\text{Cos } 2\theta}{n}$	$\frac{\text{Cos } \theta + \text{Cos } 2\theta}{n}$	$\frac{\text{Cos } 2\theta}{n}$	$\frac{\text{Cos } \theta + \text{Cos } 2\theta}{n}$	$\frac{\text{Cos } 2\theta}{n}$	$\frac{\text{Cos } \theta + \text{Cos } 2\theta}{n}$	$\frac{\text{Cos } 2\theta}{n}$	$\frac{\text{Cos } \theta + \text{Cos } 2\theta}{n}$	$\frac{\text{Cos } 2\theta}{n}$	$\frac{\text{Cos } \theta + \text{Cos } 2\theta}{n}$
0	1.0000	1.0000	0.2500	1.2500	0.2222	1.2222	0.2000	1.2000	0.1818	1.1818	0.1667	1.1667
15	0.9659	0.8660	0.2165	1.1824	0.1924	1.1583	0.1732	1.1381	0.1575	1.1270	0.1443	1.1102
30	0.8660	0.5000	0.1250	0.9910	0.1111	0.9771	0.1000	0.9660	0.0909	0.9569	0.0833	0.9493
45	0.7071	0.0000	0.0000	0.7071	0.0000	0.7071	0.0000	0.7071	0.0000	0.7071	0.0000	0.7071
60	0.5000	−0.5000	−0.1250	0.3750	−0.1111	0.3889	−0.1000	0.4000	−0.0909	0.4191	−0.0833	0.4167
75	0.2588	−0.8660	−0.2165	0.0423	−0.1924	0.0364	−0.1732	0.0856	−0.1575	0.1013	−0.1443	0.1145
90	0.0000	−1.0000	−0.2500	−0.2500	−0.2222	−0.2222	−0.2000	−0.2000	−0.1818	−0.1818	−0.1667	−0.1667
105	−0.2588	−0.8660	−0.2165	−0.4753	−0.1924	−0.4512	−0.1732	−0.4320	−0.1575	−0.4163	−0.1443	−0.4031
120	−0.5000	−0.5000	−0.1250	−0.6250	−0.1111	−0.6111	−0.1000	−0.6000	−0.0909	−0.5909	−0.0833	−0.5833
135	−0.7071	0.0000	0.0000	−0.7071	0.0000	−0.7071	0.0000	−0.7071	0.0000	−0.7071	0.0000	−0.7071
150	−0.8660	0.5000	0.1250	−0.7410	0.1111	−0.7549	0.1000	−0.7660	0.0909	−0.7751	0.0833	−0.7827
165	−0.9659	0.8660	0.2165	−0.7494	0.1924	−0.7735	0.1732	−0.7927	0.1575	−0.8084	0.1443	−0.8216
180	−1.0000	1.0000	0.2500	−0.7500	0.2222	−0.7778	0.2000	−0.8000	0.1818	−0.8182	0.1667	−0.8333

In order to get the values of F for different crank angles it is well to make a table having columns for θ, $\cos\theta$, $\cos 2\theta$, $\frac{\cos 2\theta}{n}$, $\cos\theta + \frac{\cos 2\theta}{n}$ and $\frac{WV^2}{Agr}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$.

The first five columns will have constant values for a given value of n, and values for these are given in Table XIV for five different values of n. The last column of the table, that is,

$$\frac{WV^2}{Agr}\left(\cos\theta + \frac{\cos 2\theta}{n}\right),$$

will give the values in pounds per square inch for the ordinates of the inertia diagram, the use of which is explained later.

Net Effort Diagrams. — The effect of the inertia of the reciprocating parts is to decrease the pressure at the beginning of the stroke and increase the pressure at the end of the stroke when the pressure in the cylinder has decreased due to expansion of the gas, thus tending to make the turning effort on the crank-pin more uniform.

The actual use of the inertia diagram is illustrated in Figs. 184 and 185 which are diagrams showing the combination of piston pressure with the inertia. In order to get these diagrams the lines on the indicator card are plotted in sequence, beginning with the expansion or working stroke, the total length of the diagram representing 720 degrees of the crank travel. The values of the ordinates of the inertia diagram which appear in the table as positive are plotted below the base line since they are retarding pressures, and those appearing as negative are plotted above the base line.

The pressure lines in Fig. 184 are from a single-acting, four-cycle, single-cylinder engine. The only positive piston pressure, that is, pressure which results in a positive turning moment on the crank-pin, occurs during the expansion stroke. This line will give a large area on the rotative effort diagram, the other 540 degrees of this diagram being influenced chiefly by the inertia diagram, with compression on the last stroke.

The pressure lines in Fig. 185 are from a double-acting, single-cylinder, four-cycle engine. Since the explosion strokes are but 180 degrees apart, we must expect two large areas on the rotative effort diagram close together. The effect of this will be discussed in a later paragraph. The pressure lines in Figs. 184 and

185 are taken directly from the indicator cards, the ordinates corresponding to the crank angles, say every 15 degrees. Care must be taken to get the ordinates spaced in proper order. Those

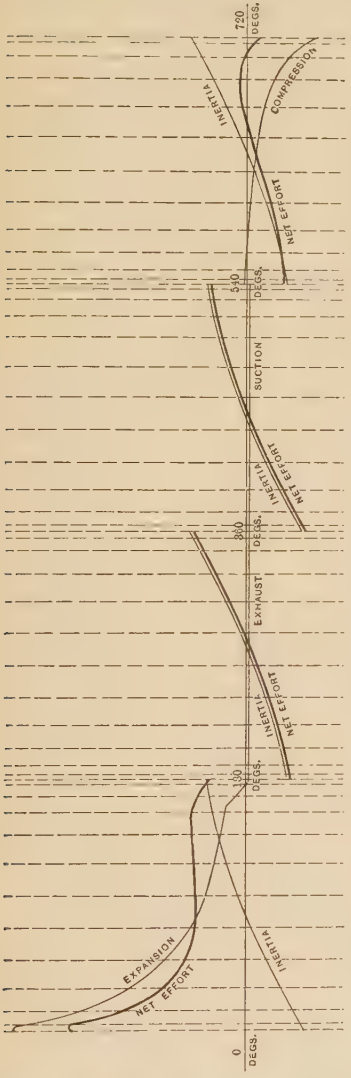


FIG. 184. — Net Effort Curve for a Four-cycle, Single-cylinder, Single-acting Gas Engine.

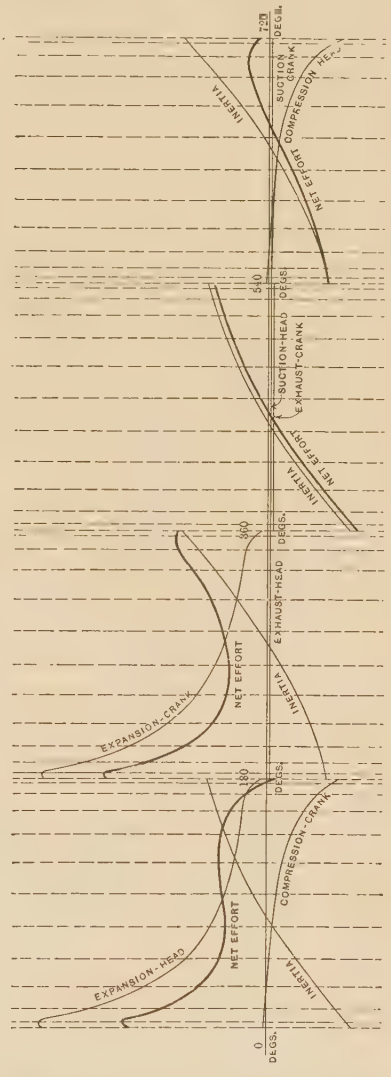


FIG. 185. — Net Effort Curve for a Four-cycle, Double-acting, Single-cylinder Gas Engine.

in the strokes from 180 to 360 degrees and from 540 to 720 degrees are in reverse order, which gives the inertia diagram the peculiar shape shown.

The **net pressure** on the piston is the difference between the pressures on the two sides of the piston occurring at the same instant. In the single-acting engine it is the difference between the pressure in the cylinder and atmospheric pressure, or it is the gauge pressure. In the double-acting engine, care must be taken to use the proper pressure as a back pressure. After the net pressure has been found, this pressure should be combined with the inertia pressure which will give the **net effort,** the pressure which is transmitted to the crank-pin. In Fig. 186 is shown a detail illustrating the manner in which these pressures should be combined. Curve 1–2 represents the expansion line for the head-end of the cylinder, 3–4 the compression line for the crank-end and 5–6 the inertia of the reciprocating parts.

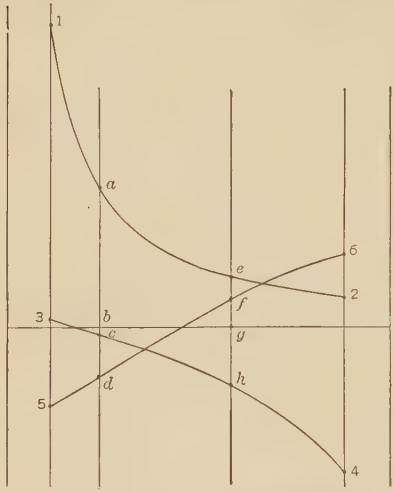


Fig. 186. — Detail of Net Effort Curve.

On the ordinate *ad* which may represent some crank angle, $ab - bc$ is the **net pressure** on the piston. The value of inertia at that point is $-bd$, so the **net effort** is $ab - bc - bd$. On the ordinate *eh*, the pressure on the expansion line is $+eg$ and that on the compression line is $-gh$, so the net pressure is $eg - gh$. The value of inertia at this point is $+fg$. Hence the net effort is $eg - gh + fg$.

If a tandem double-acting engine is under consideration there will be four pressure lines for each stroke, expansion, exhaust, suction and compression. These four may be combined as above to find the net pressure and then combined with the inertia to get the net effort.

Rotative Effort Diagram. — After the net effort on the piston has been found as in the preceding article, the rotative effort should be found. If only a few points are to be determined, a graphical construction is most convenient. Thus, in Fig. 187, if the crank is extended and *ab* is laid off equal to the net effort on piston at *c*, the intercept *cd*, found by drawing *bd* parallel to the

connecting-rod, will be the rotative effort for the crank position *oa*. *Proof:* It has been shown that when *ab* is the velocity of the crank-pin, *cd* is the velocity of the piston. Neglecting friction the work per minute will be equal at the two points. Hence the

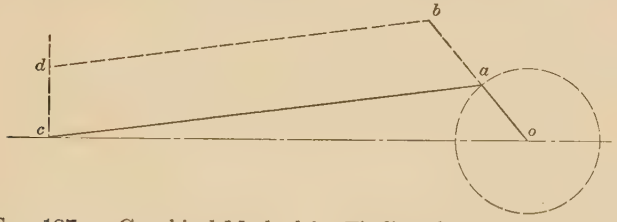


FIG. 187. — Graphical Method for Finding the Rotative Effort.

forces at the two points are inversely as the velocities, and if *cd* represents the velocity of the piston, it will also represent the working or tangential force on the crank-pin.

When a number of points are to be found, as in a rotative effort diagram, it is better to find them by calculation. In Fig. 188 we

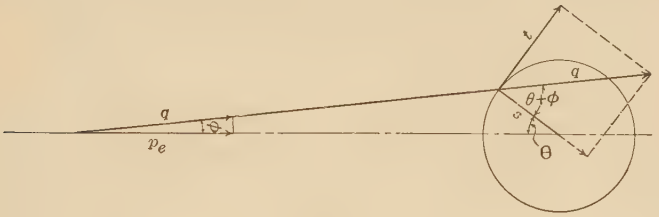


FIG. 188.

have q, the thrust along the rod, equal to $p_e \sec \phi$, where p_e is the net effort on the piston taken from the net effort diagram for that particular crank angle. Resolving q along and at right angles to the crank, we find t, the rotative or tangential effort, and s, the radial thrust.

$$q = p_e \sec \phi,$$

$$t = q \cos [90 - (\theta + \phi)] = q \sin (\theta + \phi)$$

$$= p_e \sec \phi \sin (\theta + \phi) = p_e \frac{\sin (\theta + \phi)}{\cos \phi}, \qquad (1)$$

$$\sin (\theta + \phi) = \sin \theta \cos \phi + \cos \theta \sin \phi.$$

Then $$\frac{\sin (\theta + \phi)}{\cos \phi} = \sin \theta + \frac{\cos \theta \sin \phi}{\cos \phi}. \qquad (2)$$

But $$\sin\phi = \frac{\sin\theta}{n},$$

$$\cos\phi = \sqrt{1 - \frac{\sin^2\theta}{n^2}}.$$

Hence $$\frac{\sin(\theta+\phi)}{\cos\phi} = \sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right).$$

Or $$t = p_e \sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right). \qquad (3)$$

Where t = tangential or rotative effect and p_e = pressure from net effort diagram.

In order to use equation (3) it is convenient to have a table containing the values of $\sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right)$ for different values of θ and n. The values of this expression are given in Table XV for every 15 degrees and values of n of 5, $5\frac{1}{2}$ and 6. The values in this table need only to be multiplied by the p_e taken from the net effort diagram for that crank angle. The values of t thus obtained should be used as ordinates for the curve as shown in Figs. 189 and 190, which are rotative effort curves for the net effort diagrams shown in Figs. 184 and 185.

TABLE XV

VALUES OF $\sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right)$

Degrees.	$n=4$	$n=4\frac{1}{2}$	$n=5$	$n=5\frac{1}{2}$	$n=6$
0	0.0000	0.0000	0.0000	0.0000	0.0000
15	0.3214	0.3144	0.3088	0.3043	0.3005
30	0.6091	0.5968	0.5870	0.5790	0.5724
45	0.8341	0.8196	0.8081	0.7988	0.7910
60	0.9769	0.9642	0.9540	0.9457	0.9390
75	1.0303	1.0227	1.0169	1.0121	1.0081
90	1.0000	1.0000	1.0000	1.0000	1.0000
105	0.9015	0.9091	0.9149	0.9197	0.9237
120	0.7551	0.7680	0.7781	0.7863	0.7930
135	0.5801	0.5946	0.6061	0.6154	0.6232
150	0.5181	0.4032	0.4130	0.4210	0.4276
165	0.1962	0.2032	0.2087	0.2133	0.2171
180	0.0000	0.0000	0.0000	0.0000	0.0000

The base of these curves is the distance moved through by the crank-pin during the time the piston moves through the distance

represented by the base of the net effort diagrams. In the case of four-cycle engines this should always be four strokes, hence the base of the rotative effort diagram will be $2\pi l$, where l is the length of the indicator card. When tandem, double-acting cylinders are being considered the rotative effort diagram need be only πl inches long. If the same pressure scale is used on the indicator and rotative effort diagrams, then the areas of both should be equal. This is a convenient check on the accuracy of the work at this point.

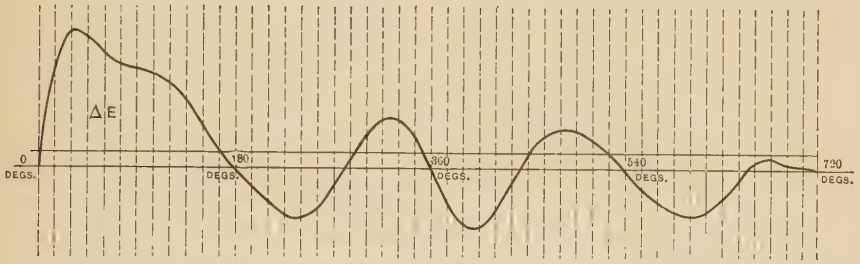


FIG. 189. — Rotative Effort Diagram for a Four-cycle, Single-cylinder, Single-acting Gas Engine.

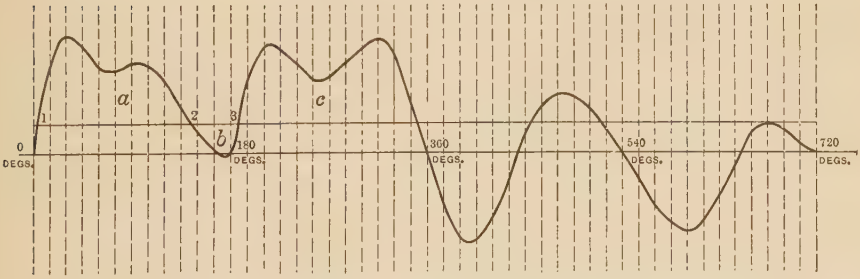


FIG. 190. — Rotative Effort Diagram for a Four-cycle, Single-cylinder, Double-acting Gas Engine.

Excess Energy. — If the area of the rotative effort diagram is found, and this area is divided by the length, the result is the mean tangential force on the crank-pin. If the actual tangential force on the crank-pin were constant there would be no need of having a flywheel. The necessity of having a flywheel is shown by the area above or below the mean tangential effort line. If we assume the power taken off the engine shaft to be constant during a cycle, then the area above or below the mean line represents the

energy absorbed and given out by the flywheel during its allowable fluctuation in speed. The maximum single area between the curve and the mean line usually is the area on which the weight of the flywheel is based. In some cases the excess area is the sum of two large areas above the mean line minus a smaller area between these two and below the line. In Fig. 189 the maximum excess area is evidently that marked ΔE. In Fig. 190 the excess area is not so evident. The flywheel increases in speed while absorbing the energy represented by a, and slows down while giving out the energy b. But as b is less than a the wheel is revolving at a higher velocity at 3 than at 1. A further increase in velocity must take place in order to absorb the energy represented by c. Hence the total energy which the flywheel must absorb during its allowable increase in speed is represented by the area $a + c - b$.

Weight of Flywheel. — The expression for kinetic energy of a body moving at a velocity of V feet per second is

$$\text{K.E.} = \tfrac{1}{2} MV^2, \tag{1}$$

where M = mass of the body. Using for M the expression $\frac{W}{g}$ in order to change to engineering units, the change in kinetic energy for a change in velocity from V_1 to V_2 will be

$$\begin{aligned}\Delta E &= \frac{W}{2g}(V_1^2 - V_2^2)\\ &= \frac{W}{2g}\frac{4\pi^2\rho^2}{3600}(N_1^2 - N_2^2)\\ &= \frac{W}{2g}\frac{4\pi^2\rho^2}{3600}(N_1 - N_2)(N_1 + N_2),\end{aligned} \tag{2}$$

where N_1 is the maximum speed and N_2 the minimum speed in revolutions per minute and ρ is the radius of gyration in feet.

$$N_1 + N_2 = 2N,$$

$$\frac{N_1 - N_2}{N} = \frac{1}{k}, \text{ the coefficient of fluctuation.}$$

Then
$$\Delta E = \frac{1}{2}\frac{W}{g}\frac{4\pi^2\rho^2 \times 1 \times 2N^2}{3600 \times k}, \tag{3}$$

when N = mean speed in revolutions per minute.

Hence
$$W = \frac{\Delta E \times g \times 900 \times k}{\pi^2\rho^2 N^2}. \tag{4}$$

The weight W thus obtained is the weight of the wheel in pounds at the radius of gyration, ρ feet.

It is usually assumed that all this weight is in the rim, or, in other words, the weight of the arms and hubs is neglected at this time, so far as their flywheel effect is concerned. The radius of gyration may be found by assuming a proper rim velocity. This should be kept below 6000 on large engines and below 5000 on small ones. After the weight is found, the rim dimensions may be computed from the formula

$$2 \times 12\,\pi\rho \times a \times b \times 0.26 = W,$$

where b = width of rim,

a = thickness of rim, both in inches.

The ratio of $\frac{a}{b}$ should be assumed and used with above equation. After a has been found the outer and inner diameters will be found to be $2\,\rho + a$ for the outer and $2\,\rho - a$ for the inner.

Wheels that are to serve as belt-wheels will naturally have wider and thinner rims than plain flywheels. The width of belt should be determined, then the width b made about 1 inch greater.

Coefficient of Fluctuation. — The value of k to be assumed in the expression $\frac{N_1 - N_2}{N} = \frac{1}{k}$ depends on the nature of the work the engine is to do. In general it might be said that engines that drive through belts do not require so large a flywheel as engines that are direct-connected to the machine they drive. The following values of k are taken from Levin's "Modern Gas Engine and the Gas Producer."

	k
For ordinary industrial purposes, belt-driven	25– 35
For small electric light installations, direct-current, belt-driven	50– 60
For pumping machinery, direct-connected to engine	60–100
For large electric light installations, direct-current, belt-driven	60– 80
For large electric light installations, direct-connected	90–120
For gear-wheel transmissions	90–120
For blast-engines	90–150

Haeder and Huskisson, in their "Hand Book on the Gas Engine" give the following values:

For direct-current, direct-connected running singly	120
For direct-current, direct-connected running in parallel	200

For alternating-current, direct-connected running singly...	200
For alternating-current, direct-connected running in parallel	250
For direct-current, belt-connected, running singly.....	80
For direct-current, belt-connected, running in parallel.....	120
For alternating-current, belt-connected, running singly....	100
For alternating-current, belt-connected running in parallel.	180

Construction Details. — Flywheels up to 8 feet in diameter may be cast solid, but all above this diameter should be cast in halves or smaller sectors. The material is usually cast-iron. In cases where extremely high speed is necessary the wheel may be made of steel.

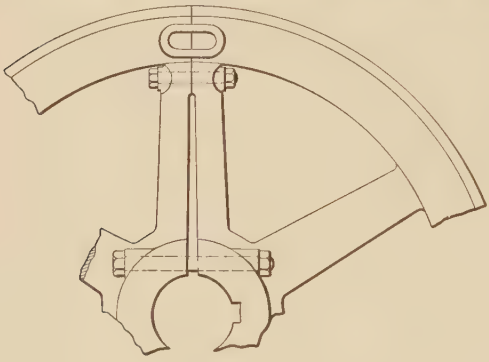


Fig. 191.

The simplest form of built-up wheel is the one cast in halves. In this case the parting should be made on an arm and not between the arms. Fig. 191 shows one manner of fastening the halves together. In addition to the hub bolts which clamp the

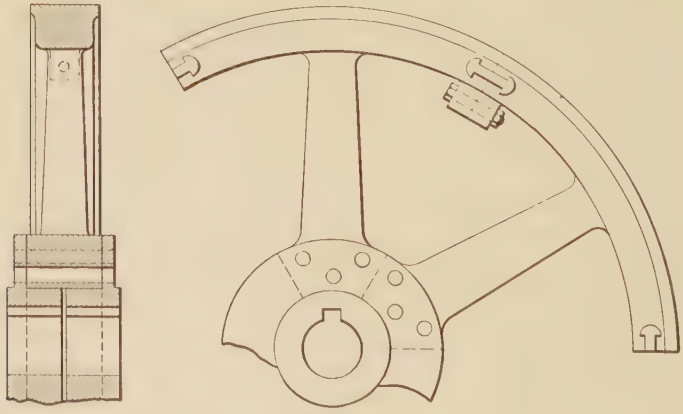


Fig. 192.

wheel on the shaft, there should be one or two keys. The total area of the bolts and shrink links at the rim should be at least one-fifth the sectional area of the rim.

When the wheel is built in sectors, usually six pieces, each arm is cast with a portion of the rim. The pieces of rims are held together by shrink links and bolts and the inner ends of the arms are bolted between two flanges at the hub. This method is illustrated in Fig. 192.

Tension in Rim. — The tension in the rim due to the centrifugal force may be found as follows:

$$\frac{awV^2}{gR} = \text{centrifugal force of 1 inch of rim,} \qquad (1)$$

when a = sectional area of rim in square inches,
w = weight of metal in pounds per cubic inch,
V = velocity of rim in feet per second,
R = maximum radius of rim in feet,
$R = \frac{D}{24}$, when D is the diameter in inches.

Then the centrifugal force becomes $\frac{24\,awV^2}{gD} = \frac{3\,awV^2}{4\,D}$.

The resultant force on one side of a diameter will be $D \times \frac{3\,awV^2}{4\,D}$ and this force will be resisted by $2\,Ka$, the tension in both sides of the rim. Then

$$2\,Ka = D \times \frac{3\,awV^2}{4\,D}, \qquad (2)$$

$$K = \frac{3\,wV^2}{8}. \qquad (3)$$

Design of the Arms. — Flywheel arms are usually made elliptical in section, with the major axis twice the minor. The usual number of arms is six. In very heavy flywheels eight arms are sometimes used.

The arms should be made strong enough to withstand the maximum twisting moment which it is possible for them to receive from the piston. This is approximately $p \times A \times r$, where

p = maximum pressure per square inch on piston,
A = area piston in square inches,
r = crank radius in inches.

This twisting moment will be equally distributed among all the arms, hence the bending moment on one arm will be

$$\frac{p \times A \times r}{n} \quad \text{where} \quad n = \text{number of arms.}$$

The section modulus of an ellipse is $\frac{\pi ab^2}{32}$, or $\frac{\pi b^3}{64}$, when a, the minor axis, is one-half the major. Then

$$\frac{p \times A \times r}{n} = \frac{K\pi b^3}{64}, \tag{1}$$

$$b = \sqrt[3]{\frac{P \times A \times r \times 64}{\pi n K}}. \tag{2}$$

The stress K may be assumed to be 2000 when used under the conditions mentioned above. The value of b found by (2) would be at the center of the shaft. The dimension b' at the rim may be found in several ways, one of which is to assume the taper to be $\frac{3}{8}$ inch per foot over all. Another way is to assume b' to be $\frac{2}{3}\,b$. The minor axis a' at the rim may be found by either method. If the taper is used it should be limited to $\frac{3}{16}$ inch per foot. To find the dimensions of the arms at the hub when the drawing of the flywheel is made, use the taper as given above.

After the dimensions of the arms at the hub have been determined, the stress on this section, due to the centrifugal force of $\frac{1}{n}$th of the rim, should be determined. To this should be added the tension due to the weight of one-half the rim, since an arm in the bottom vertical position is assumed to carry the weight of half the rim. The total stress in an arm at the hub, when the flywheel is up to speed will be $K_c + K_w$, where

K_c = stress due to centrifugal force of rim,
K_w = stress due to weight of one-half the rim.

Example. — Design the flywheel arms for a 16 × 24-inch gas engine, maximum pressure in the cylinder 400 pounds per square inch. Weight of rim 10,600 pounds, diameter 9 feet. Velocity of rim 5600 feet per minute.

Assume 6 arms.

$$M_b = \frac{400 \times 201 \times 12}{6} = 160{,}800 \text{ inch-pounds},$$

from relation on page 327.

From (2) we get

$$b = \sqrt[3]{\frac{160{,}800 \times 64}{\pi \times 2000}} = 11\tfrac{7}{8} \text{ inches}.$$

If we assume the radius out to the junction of the arms to be

1 foot, the width of section there will be $11\frac{1}{2}$ inches, thickness $5\frac{3}{4}$ inches.

The centrifugal force of one-sixth of the rim will be

$$\frac{10{,}600}{6} \times \frac{5600 \times 5600}{3600 \times 32 \times 4.5} = 109{,}000 \text{ pounds.}$$

$$K_c = \frac{109{,}000}{\frac{\pi}{4} \times 11.5 \times 5.75} = 2092 \text{ pounds per square inch.}$$

$$K_w = \frac{10{,}600}{2 \times \frac{\pi}{4} \times 11.5 \times 5.75} = 102 \text{ pounds per square inch.}$$

Total direct tension = 2092 + 102 = 2194 pounds per square inch. This stress should not exceed 2500 pounds per square inch.

Stress in Arms Due to Inertia of Rim. — The design of the flywheel arms as given in a previous article takes into account the total piston pressure and hence will give a certain size of flywheel arms for a given size of engine. In other words, the size of the arms will be independent of the size of the rim. In some cases arms designed in the manner given above will be weak. In any case where a large load is thrown on the engine suddenly, slowing the engine down, the inertia of the flywheel tends to keep the engine up to speed and the force of the inertia bends the flywheel arms. One very good illustration is the engine driving a rolling-mill. As the bar of steel strikes the rolls the load on the engine is increased instantly from zero to a maximum. The flywheel then keeps the engine going until the governor opens the valves. Gas engines are rarely used on rolling mills but the same effect might be secured in an engine driving plunger pumps, or even in engines driving electric generators.

The arms of a flywheel should be tested to see what stress will result when the engine stops quickly. The weak point in this step is that the time of stopping must be assumed. After the time is assumed the force necessary to stop the wheel in that time may be found from $F = Ma = \frac{Wa}{g}$, where F = force in pounds, W = weight of rim in pounds and a = negative acceleration in feet per second per second. The force F will be applied out at the center of gravity of the rim and will bend the arms at the hub with a moment Fl, where l is the distance from the center of gravity of the rim to the hub.

Example. — The flywheel used in the last example had a rim 9 feet in diameter, weight of rim 10,600 pounds, velocity of rim 5600 feet per minute or 93.3 feet per second. This velocity corresponds to 200 r.p.m. If we assume the flywheel to make two revolutions in coming to rest, the time required will be

$$\frac{2 \times 9 \times \pi \times 2}{93.3} = 1.2 \text{ seconds.}$$

The force of acceleration of the rim will then be

$$F = \frac{10{,}600 \times 93.3}{32.2 \times 1.2} = 25{,}600 \text{ pounds.}$$

We have already made the assumption of the distance from the center of the wheel to the junction of the arms to be 12 inches. If we make the further assumption that the rim is 4 inches thick, the lever arm of the force F will be 40 inches. The bending moment on one arm will then be

$$M = \frac{25{,}600 \times 40}{6} = 170{,}500 \text{ inch-pounds,}$$

and the stress will be

$$K = \frac{170{,}500 \times 64}{11\frac{1}{2}^3 \times \pi} = 2280 \text{ pounds per square inch.}$$

Velocity and Displacement Diagrams.—Let it be assumed that the length of the rotative effort diagram for the single-cylinder engine, Fig. 189, represents the time it takes the flywheel to make two revolutions instead of representing the distance traveled by the crank-pin in one cycle or two revolutions. This assumption may be made without error because the difference between the intervals of time and intervals of space is so small that it could not be measured on the diagram.

The driving effort acting tangentially on the crank-pin is represented for all crank positions by the ordinates of the crank-effort curve, but by multiplying this force, measured at any point of the cycle, by the quantity $\frac{g}{W}$ we obtain the acceleration which the tangential effort gives a weight W. The tangential effort curve can, therefore, be said to be the acceleration curve for a weight W.

The distance between successive ordinates of the diagram represents the time required for $\frac{1}{24}$ of a revolution. The height of the

mean ordinate between the base-line and the acceleration curve during any interval represents the average of the variable acceleration which a weight W is given during the time $\frac{1 \times 60}{24 \times N}$ seconds, $\frac{60}{N}$ being the time for one revolution.

The change in the velocity of a moving object during a given period of time is the product of its acceleration and the time. Hence, the area of the curve, Fig. 189, which we may now call the acceleration curve, up to any given ordinate, represents the velocity at that point or, in other words, the area included between any two ordinates, the curve and the mean effort line represents the change in velocity during the time represented by the distance between the ordinates.

In order to represent graphically the increase or decrease in the velocity of the revolving parts of the engine we may, therefore, integrate the elementary areas beginning at any point of the cycle and plot their sum on corresponding ordinates, having as a base-line the base of the acceleration curve. In Fig. 193 is shown

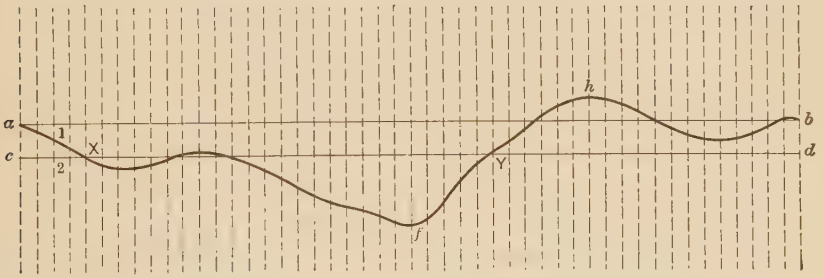


FIG. 193. — Velocity Diagram for a Four-cycle, Single-cylinder, Single-acting Gas Engine.

such a velocity curve plotted from the rotative effort or acceleration curve in Fig. 189. The velocity curve was started at the point on the acceleration curve corresponding to the beginning of the suction stroke.

The line *ab* on Fig. 193 is the base-line and *cd* the mean velocity line, the distance between *ab* and *cd* being the area between the curve and base-line divided by the length of the curve.

The vertical distance between the low point of the curve *f* and the high point *h* represents the maximum change in velocity of

the flywheel during the time represented by the horizontal distance between f and h. During this time the flywheel absorbs or gives out the energy represented by ΔE in Fig. 189.

The area between the line cd, the **velocity curve** and any two ordinates is the product of the excess or deficiency in velocity and the time. Hence we may integrate the areas curve and cd, the line of normal velocity, and the area thus found up to any point will be proportional to the displacement of a point on the flywheel at that time. If the velocity of the revolving system is constant, then each point of the system will be in its proper position each instant, but as the velocity varies from the normal, so will the position of a given point vary from the normal and the variation is shown by the ordinates of Fig. 194, which is the **displacement curve.**

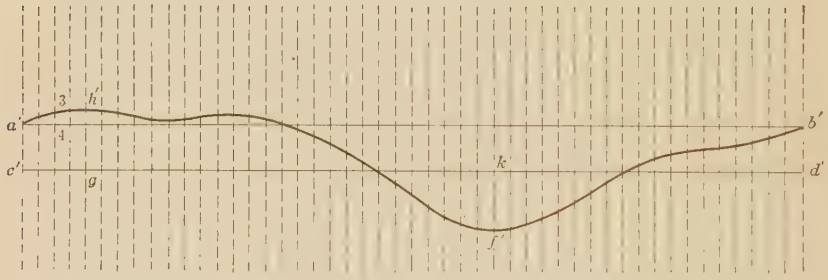


FIG. 194. — Displacement Diagram for a Four-cycle, Single-cylinder, Single-acting Gas Engine.

The line $a'b'$ of Fig. 194 is the base-line from which the ordinates are set off, representing areas of the velocity curve up to the corresponding ordinate on the velocity curve. Thus, the ordinate 3–4 of the displacement curve is proportional to the area a 12 c of the velocity curve and is laid off in the positive direction since the area a 12 c is above the mean line and hence positive.

Since a point in the revolving system will continue to fall further behind its normal position so long as the velocity is below normal, it follows that the point of maximum displacement ahead of normal will be at the end of a period of velocity above the normal. Thus the point where the velocity curve crosses the mean velocity line at X is the end of a positive velocity area. Hence the maximum ahead displacement h' will occur directly under X on the velocity curve. Likewise the maximum displacement in the oppo-

site direction will occur at f', directly under Y, the point at which the velocity curve crosses the mean velocity line after a period of low velocity.

The area of the velocity curve above the mean velocity line is equal to the area below the mean velocity line. If the velocity were constant, then there would be no displacement ahead or behind the normal. Since the displacement is proportional to the area of the velocity curve, it follows that a revolving point will get a certain distance ahead of its normal position during a revolution and will fall the same distance behind, but at some point, or at two points in the revolution, it will be in its normal position. Hence, if we draw a horizontal line $c'd'$, Fig. 194, in such a way that gh' equals $f'k$, then the ordinates from this line to the curve will be proportional to the distance the flywheel is ahead of or behind its normal position at the time corresponding to the position of the ordinate.

Application of the Velocity and Displacement Diagrams. — Let us assume a condition which places an alternating-current generator driven by a water-wheel in parallel operation with one driven by a gas engine, the generator in each case to be direct-connected to its prime mover. Let us also assume the power taken from the generators during a period of time corresponding to a revolution to be constant. Then the generator driven by the water-wheel will revolve at a constant velocity but the one driven by the gas engine will get ahead of and lag behind the other an amount proportional to the maximum displacements as shown by the displacement diagram. The position of the gas engine-driven generator at any time relative to the water-wheel-driven generator is proportional to the ordinate of the displacement curve at that instant.

The actual amount which the gas engine-driven generator gets behind or ahead of the other depends on the combined flywheel effect of the flywheel and generator. The flywheel formula may be written

$$W = \frac{2\,g\Delta E}{2\,V\,(V_1 - V_2)} = \frac{2\,g\Delta E}{4\,\pi\rho N\,(V_1 - V_2)},$$

where W is the combined weight of flywheel and rotor reduced to a common radius of gyration ρ. The weight increases as $(V_1 - V_2)$ decreases, and consequently as the displacement decreases. The weight, then, is dependent on the allowable dis-

placement. This allowable displacement is usually given as $2\frac{1}{2}$ degrees phase, or it is $\frac{2.5}{360} = 0.0069$ of the distance from one positive pole to the next positive pole. If the generator is a two-pole machine, then the displacement on the flywheel rim will be $2\frac{1}{2}$ degrees. If n = the number of poles on the generator, $\frac{2.5 \times 2}{n}$ will be the degrees displacement on the flywheel circle. If given in radians the angle will be $\frac{2.5 \times 2 \times \pi}{180\, n}$. The relation between this angle and the displacement curve is given here.

In Fig. 189 it is evident that the part above the mean line marked ΔE is the excess energy. Let F be the height of the curve above the mean line at any point within the area ΔE. Then the excess torque at this point is

$$FRAS = \frac{W\rho^2}{g}\frac{d^2\theta}{dt^2}, \tag{1}$$

where R = radius of crank in feet,
A = area piston in square inches,
F = height of curve above mean line in inches at any point within the area ΔE.
S = scale of curve, pounds per inch,
W = weight of flywheel and generator rotor reduced to ρ feet,
where ρ = radius of gyration.

If we multiply both sides of equation (1) by dt and change $\frac{d^2\theta}{dt^2}$ we get

$$FRAS\, dt = \frac{W\rho^2}{g}\frac{d}{dt}\left(\frac{d\theta}{dt}\right)dt. \tag{2}$$

Since the length of the curve in Fig. 185 corresponds to 2 revolutions, we have for the time in seconds, per inch of curve,

$$t = \frac{2 \times 60}{lN}, \tag{3}$$

where l = length of curve in inches and N = r.p.m. The time dt required for a differential length dx will be

$$dt = \frac{2 \times 60}{lN}dx. \tag{4}$$

Substituting this value of dt in (2) and integrating,

$$\frac{RASg}{W\rho^2}\frac{120}{lN}\int F\, dx = \int \frac{d}{dt}\frac{d\theta}{dt}\, dt = \frac{d\theta}{dt}. \tag{5}$$

Let $\frac{RASg}{W\rho^2}\frac{120}{lN} = C$. $\int F\,dx$ is the area enclosed by the rotative effort curve and the mean line up to any point and $\frac{d\theta}{dt}$ is the angular velocity of that flywheel at that point. When $\int F\,dx$ up to any point is multiplied by C, the product will be the angular velocity of the flywheel at that instant in radians per second. Then (5) becomes

$$C \times \text{area acceleration curve to any point} = \frac{d\theta}{dt} = \omega. \qquad (6)$$

When the velocity curve is constructed from the acceleration curve, a scale must be selected. Thus, 1 inch on the ordinates of the velocity curve will represent a certain number of square inches of the area of the acceleration curve. Call this scale S_1. Let h_1 = the height of the velocity diagram above or below the mean line at any point. Then in (6) we may substitute $h_1 \times S_1$ for the area of the acceleration curve to any point. If that is done and if both sides of the equation are multiplied by dt and the equation integrated, the result will be

$$C \times S_1 \int h_1\,dt = \int \frac{d\theta}{dt}\,dt. \qquad (7)$$

Substituting for dt its value from (4), we get

$$C \times S_1 \times \frac{120}{lN} \int h_1\,dx = \theta, \qquad (8)$$

where θ is the angular displacement in radians, since $\int h_1\,dx$ is the area of the velocity curve up to any point.

If the scale on the displacement curve is S_2 square inches of the velocity curve per inch of height h_2, then for $\int h_1\,dx$ in (8) we may put S_2h_2 giving

$$CS_1S_2 \times \frac{120}{lN} \times h_2 = \theta. \qquad (9)$$

Equation (9) may be used to find the weight of the flywheel for a given value of θ or it may be used to find the angular displacement θ for a flywheel determined by some other method. If the first method is used the value of θ should be found from the allowable fluctuation in degrees of phase. Thus, if the allowable

fluctuation is α degrees of phase and n is the number of poles on the alternator,

$$\theta = \frac{\alpha \times 2 \times \pi}{180\, n}. \tag{10}$$

If equation (9) is used to find the displacement with a flywheel of a given weight, the value of θ thus found should be less than that indicated by equation (10).

Approximate Weight of Flywheel. — It is sometimes desirable to find the approximate weight of a flywheel without finding the rotative effort diagram for a given engine. The excess energy ΔE which must be absorbed and given out by the flywheel bears some relation to the work generated per revolution or per cycle in the engine cylinder. This ratio

$$\frac{\text{Energy absorbed by flywheel}}{\text{Indicated work per revolution}}$$

is practically constant for a given type of engine.

The author has found the following values for the ratio given above:

TABLE XVI

FLYWHEEL CONSTANTS FOR APPROXIMATE CALCULATION

Type of engine, four-cycle.	$\Delta E \div$ indicated work per revolution.
Single-acting:	
Single-cylinder	2.35–2.40
2-cylinder twin, cranks at 360 degrees	0.92–1.04
2-cylinder twin, cranks at 180 degrees	1.50–1.60
2-cylinder opposed, cranks at 360 degrees	1.50–1.60
2-cylinder opposed, cranks at 180 degrees	0.92–1.04
2-cylinder tandem	0.92–1.04
3-cylinder vertical, cranks at 120 degrees	0.60–0.75
4-cylinder vertical, crank at 90 degrees	0.18–0.20
Double-acting:	
Single-cylinder	1.50–1.60
2-cylinder tandem	0.18–0.20
2-cylinder twin	0.18–0.20
2-cylinder twin-tandem, cranks at 90 degrees	0.07–0.075

Example. — Find the weight of flywheel for a 1000-I.H.P. 2-cylinder tandem double-acting gas engine running 85 r.p.m. Assume all the weight to be in the rim. Coefficient of fluctuation is $\frac{1}{120}$. Diameter of wheel 20 feet.

$$\Delta E = \frac{0.2 \times 1000 \times 33{,}000}{85} = 78{,}000 \text{ foot-pounds.}$$

From equation (4), page 320,

$$W = \frac{\Delta E \times g \times 900 \times k}{\pi^2 \rho^2 N^2}$$

$$= \frac{78{,}000 \times 32 \times 900 \times 120}{\pi^2 \times 10 \times 10 \times 85 \times 85} = 37{,}300 \text{ pounds.}$$

EXERCISES

1. A cast-iron flywheel rim weighs 22,000 lbs. and is 13 ft. mean diameter. Find dimensions of rim if section of rim is square.

2. Given: $\triangle E$ = 38,300 ft.-lbs., radius of gyration $7\frac{1}{2}$ ft., r.p.m. 160. Find weight of flywheel rim for a gas engine direct-connected to a direct-current dynamo running singly.

3. Find size of arms for a flywheel 14 ft. dia. for a 26 × 34-in. engine, explosion pressure 400 lbs. Outer radius of flywheel hub 14 ins. Use six arms of elliptical section, major axis of ellipse twice the minor.

4. A flywheel rim weighs 33,000 lbs. and has six arms of elliptical section, 14 × 7 ins. Distance from center of gravity of rim to hub 4′–3″. Find stress in arms if flywheel comes to rest in two revolutions, from speed of 165 r.p.m.

5. Find weight of flywheel rim for 90 horse-power single-cylinder, single-acting, four-cycle engine running 160 r.p.m., wheel to have radius of gyration of 4′–6″. Assume k = 60.

6. Design the arms for flywheel in Prob. 5. Find total weight of wheel.

CHAPTER XXI

CYLINDERS — CYLINDER COVERS — FRAMES — VALVES — VALVE-GEARS

Cylinders. — The **materials** used in gas engine cylinders are cast-iron and cast-steel. If iron is used it should be hard and close-grained. Steel is used only where cast-iron liners are used. Allowable stress is fairly high so that the walls may be thinner than if a lower stress is used. With thinner walls conduction of heat is more rapid than with thick walls and stresses due to heating up are lower. For cast-iron with clearly defined load conditions, the allowable stress in tension may run as high as 4000 pounds per square inch.

Cylinders and jackets are often cast in one piece and in very small engines the cylinder, jacket and frame may be combined in one casting, although this is not the usual practice. In some instances where the cylinder and jacket are cast as one piece, they are connected at one end only and are free from each other at the other end. This arrangement is shown clearly in Fig. 195 which illustrates the cylinder of a vertical Diesel engine. The liner and cylinder are connected at the top but left free at the bottom where the opening thus formed in the water-space is filled by a ring held in place by means of cap-screws. The same method is sometimes used in double-acting cylinders. In this case the jacket is cast integral with the cylinder but is not continuous from end to end. A portion is left out at the middle of the cylinder and the opening thus formed is closed by a separate steel band. Rubber rings are laid in U-shaped grooves at the ends of the jacket and the bands are pulled up on these rings, forming a water-tight joint. This method is illustrated in Fig. 196. Figs. 69 and 81 illustrate clearly the practice of casting the cylinder and jackets in one piece without leaving the jacket free at one end. Fig. 126 illustrates an engine in which a separate liner has been used, the liner forming the cylinder proper and the space between the liner and jacket the water-space. A separate liner has the advantage that a specially suitable grade of iron can be employed and the jacket

and frame may be cast with greater ease. It has the additional advantage that the wear on the cylinder may be taken care of

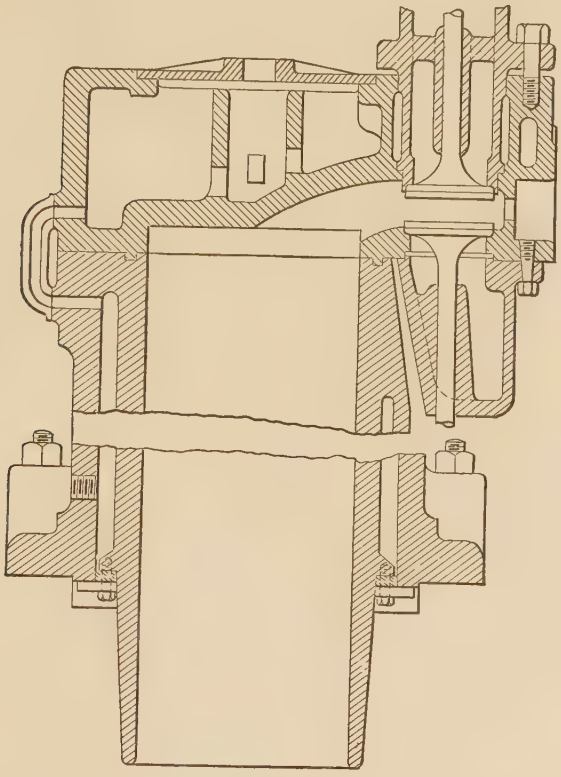


FIG. 195.—Cylinder for Vertical Multicylinder Gas Engine or Diesel Engine.

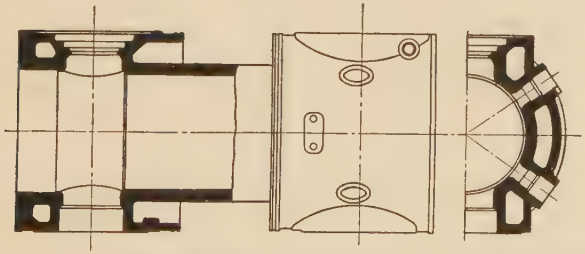


FIG. 196. — Double-acting Gas Engine Cylinder with Non-continuous Jacket.

more readily. A further advantage, one that is very important, is that expansion due to high temperature in the cylinder is taken

care of without imposing dangerous stresses on cylinder and jacket. To allow this free expansion, the liner is held rigidly at one end while the other is allowed to move axially.

Constructive Details. — The thickness of the cylinder should be determined to withstand the internal pressure of the explosion

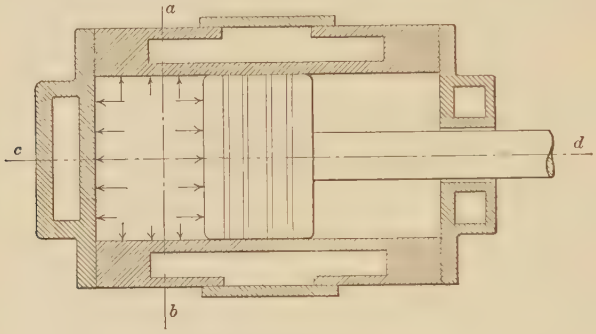


Fig. 197.

and the stresses brought on it by the connecting-rod thrust in a trunk-piston engine. The internal pressure stresses the cylinder barrel in a longitudinal direction and in a radial direction as shown by Fig. 197. The cylinder wall proper takes all the radial load and all the axial load if the water jacket is not continuous and cast integral with the cylinder.

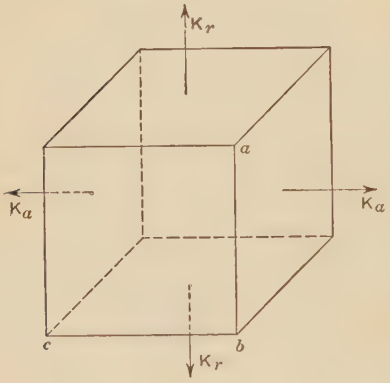


Fig. 198.

Let K_a be the unit stress due to the axial load and K_r due to the radial load. Then a unit cube of metal in the cylinder wall will be stressed as shown in Fig. 198. The stress K_r tends to increase the length of ab and decrease cb, while K_a has the opposite effect, tending to increase cb and decrease ab. From the stress strain relation we have, in general,

$$\frac{K}{e} = E, \quad \text{and} \quad K = Ee, \tag{1}$$

where K is the unit stress, e the unit strain or deformation and E

is the modulus of elasticity. The relation between the elongation of ab and compression of cb due to K_r is given by $\frac{1}{m}$, Poisson's ratio. That is, the decrease in the length cb due to K_r is $\frac{1}{m}\frac{K_r}{E}$ and the decrease in the length ab due to K_a is $\frac{1}{m}\frac{K_a}{E}$. Hence the net deformation of ab is

$$e_1 = \frac{K_r}{E} - \frac{1}{m}\frac{K_a}{E}. \tag{2}$$

Using (2) and (1) we may find the net stress K_1 due to the net deformation e_1.

$$K_1 = Ee_1 = K_r - \frac{1}{m}K_a. \tag{3}$$

Likewise the stress in direction of cb is

$$K_2 = Ee_2 = K_a - \frac{1}{m}K_r. \tag{4}$$

The thickness of the cylinder wall may be found by using the formula for a thin cylinder, neglecting the stress due to the pressure on the ends of the cylinder,

$$t = \frac{pD}{2K_r} + C, \tag{5}$$

where p is the internal pressure in pounds per square inch, D the diameter in inches, K_r the allowable fibre stress and C a constant to allow for reboring. The constant depends on the diameter of the cylinder and may be selected from Table XVII.

TABLE XVII

VALUES OF C

Diameter of cylinder	4	6	8	10	12	14	16	18	20	24	28	32	40	48
C in inches	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$1\frac{1}{16}$	$1\frac{3}{16}$

Example. — Find the thickness of cylinder walls for a 12-inch gas engine cylinder, explosion pressure 450 pounds per square inch.

$$t = \frac{450 \times 12}{2 \times 4000} + \frac{1}{2} = 1\tfrac{3}{16} \text{ inches.}$$

The stress due to the pressure on the ends of the cylinder may be found by the formula

$$\frac{\pi}{4}(D_1^2 - D^2)K_a = \frac{\pi D^2}{4}p, \tag{6}$$

or
$$K_a = \frac{D^2}{D_1^2 - D^2} p, \tag{7}$$

where D is the inside and D_1 the outside diameter of the cylinder in inches. In the problem above $D = 12$ inches and $D_1 = 14\frac{3}{8}$ inches. Hence

$$K_a = \frac{\overline{12}^2}{\overline{14.375}^2 - \overline{12}^2} \times 450 = 1035 \text{ pounds.}$$

The stress due to radial pressure will be

$$K_r = \frac{450 \times 12}{2 \times 1.18} = 2290 \text{ pounds.}$$

From (3) we find the stress due to the combination of K_a and K_r to be

$$K = 2290 - \frac{1}{4} \times 1035 = 2030 \text{ pounds,}$$

where $\frac{1}{4} = \frac{1}{m}$ for cast iron. Since the resulting stress is almost as large as the maximum stress, the maximum is used in cylinder design. Concerning the above problem the stresses will increase greatly after the cylinder is rebored and care should be taken that reboring does not increase the stresses until they become dangerously high.

Some designers of gas engines use the formula for thick cylinders for finding the thickness of the cylinder wall. The one used by Güldner in his "Design and Construction of Internal Combustion Engines" is

$$t = \frac{D}{2}\left(\sqrt{\frac{K + 0.4\,p}{K - 1.3\,p}} - 1\right) + C \text{ inches.} \tag{8}$$

Using the same example as before we find

$$t = \frac{12}{2}\left(\sqrt{\frac{4000 + 0.4 \times 450}{4000 - 1.3 \times 450}} - 1\right) + \frac{1}{2} = 1.142 \text{ inches.}$$

Due to the internal pressure the inner fibres of the cylinder wall are under a greater stress than the outer fibres. This inequality of stress increases materially with an increase in the thickness of the wall. For that reason t should not be made greater than is necessary. Excessive thickness is also wrong from the standpoint that the cooling water is less effective, consequently the conduction of heat is less rapid, and troublesome overheating of the material may ensue.

Stress in Cylinder Walls Due to Connecting-rod Thrust. — In large or long-stroke engines the thrust of the connecting-rod should receive careful attention in finding the thickness of the cylinder. In case the formulæ given should furnish a section modulus too small to take care of the normal pressure T due to the connecting-rod, it is better to increase the stiffness of the cylinder wall by suitable supports or webbing rather than to increase the thickness.

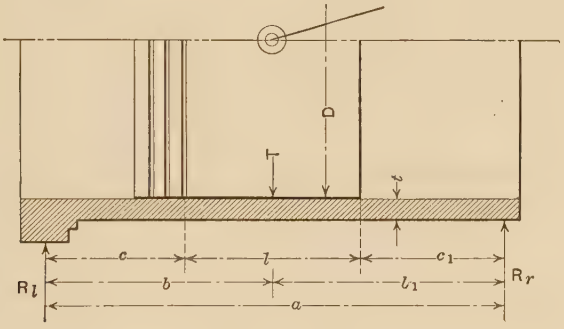


FIG. 199.

With the favorable assumption that T is distributed over the bearing surface of the length l, Fig. 195, the maximum bending moment will be

$$M = R_l\left(c + \frac{lR_l}{2\,T}\right) \leqq \frac{K_b I}{d_1}, \tag{1}$$

in which

$$R_l = \frac{T\,(2\,c_1 + l)}{2\,a} \tag{2}$$

and $\dfrac{I}{d}$ = section modulus.

The stiffness of the cylinder liner or the cylinder should be such that the vertical component T cannot cause any noticeable bending or deformation. In order to make sure of this point it is advisable to make a check computation, assuming that the force T, instead of being distributed over the length l, is concentrated at the middle of this length. In that case

$$M = T\frac{bb_1}{a} \text{ inch-pounds}, \tag{3}$$

and assuming that K_b is the allowable stress, the section modulus required will be

$$\frac{I}{d_1} = T\frac{bb_1}{aK_b} \geqq \frac{M}{K_b}. \tag{4}$$

The deflection of the cylinder will be

$$\delta = \frac{Tb^2b_1^2}{3\,aIE}.$$

Example. — A cylinder liner with a diameter of 25 inches and an explosion pressure of 400 pounds per square inch will have a thickness of

$$t = \frac{400 \times 25}{2 \times 4000} + \frac{9}{16} = 1\tfrac{3}{16} \text{ inches.}$$

With reference to Fig. 195 let $a = 50$ inches, $b = 24$ inches, $b_1 = 26$ inches. Then the maximum value of T will be approximately

$$T = 0.10 \times \frac{\pi \times 25^2}{4} \times 400 = 19{,}640 \text{ pounds.}$$

Then $$M_b = \frac{19{,}640 \times 24 \times 26}{50} = 245{,}000 \text{ inch-pounds.}$$

$$\frac{I}{d} = \frac{\pi}{4}D_m^2 t = 0.7854 \times 26.81^2 \times 1.81 = 1023,$$

where D_m is the mean diameter of the cylinder barrel. The maximum bending stress therefore is

$$K_b = \frac{245{,}000}{1023} = 240 \text{ pounds per square inch.}$$

This stress is low and apparently the cylinder is amply strong but the maximum deflection should be determined. The moment of inertia of the cross-section is

$$I = \frac{\pi D_m^3 t}{8} = \frac{\pi \times 26.81^3 \times 1.81}{8} = 13{,}700.$$

Using a value of 12,000,000 for E for cast-iron we find the maximum deflection to be

$$\delta = \frac{19{,}640 \times 24^2 \times 26^2}{3 \times 50 \times 13{,}700 \times 12{,}000{,}000} = 0.000315 \text{ inch.}$$

Since the load T is not concentrated but is distributed over the length l which makes δ still smaller, the result shows that this cylinder liner is safe without any further stiffening.

Cylinder Flanges. — In many cases gas engine cylinders do not have flanges but the studs which retain the head in place are screwed into the metal joining the cylinder wall and jacket wall. This condition is clearly shown in Figs. 195 and 196. In cases where it is necessary to use a flange this flange should be made thicker than the cylinder wall and the depth of the flange a, Fig. 200, should be made as small as possible, just large enough to accommodate the bolts. The stress on the bolts due to drawing up the packing at b combined with the explosion pressure tends to break the flange along the line bd. This necessitates making t' about 1.25 t, where t is the thickness of the cylinder wall, and also keeping a as small as possible. In addition a generous fillet should be used at the junction of the flange and cylinder wall.

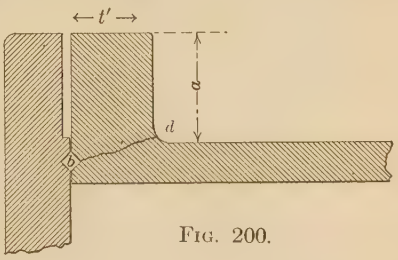


Fig. 200.

Jacket Wall. — The stress on the jacket wall depends entirely on the arrangement of cylinder and jacket walls. When the cylinder and jacket are cast in one piece, as in Fig. 201a, the jacket

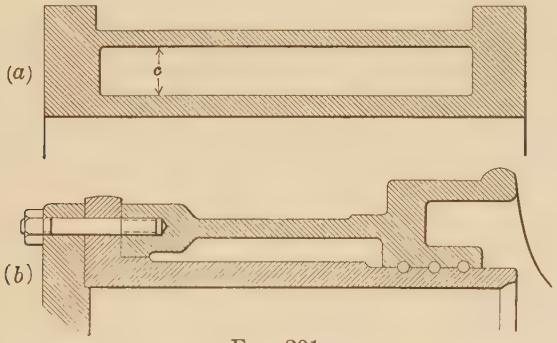


Fig. 201.

takes part of the axial thrust due to the explosion pressure. If the arrangement is as shown in Fig. 201b the jacket takes all the axial thrust and should be designed accordingly. In small engines sheet-iron or copper jackets are sometimes used. Where they are used the cylinder wall must be depended on to take all loads due to the explosion. In general it might be said that the thickness of the jacket wall should be from two-thirds to three-

fourths of the thickness of the cylinder wall. The depth of the water space c, Fig. 201a, should be from $\frac{3}{4}$ inch for a 6-inch cylinder to 3 inches for a 30-inch cylinder.

Cylinder Cover. — The cylinder cover is usually made of tough, close-grained cast iron. The allowable stress should, in such places where it can be determined with any accuracy, be kept moderately high, say 3500 pounds per square inch. This is done in order to get a construction that is not too rigid against expansion by heat and to obtain better conduction of heat through the walls. In general, however, the less the certainty with which the forces acting can be determined, the lower should be the value of the stress.

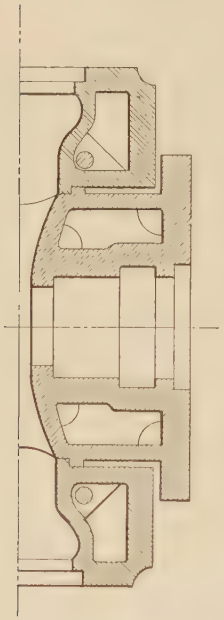


Fig. 202. — Cylinder Cover for a Double-acting Gas Engine.

In cases where the cylinder cover is a flat plate the thickness could be found from the formula

$$t = D\sqrt{\frac{cp}{K}},$$

where t = thickness in inches,
D = diameter of cylinder in inches,
p = pressure in pounds per square inch,
c is a constant and K the allowable fibre stress. The value of the constant is about 0.10 for gas engine work.

Example. — Find the thickness of a plain cylinder cover for a 12-inch cylinder, pressure 450 pounds.

$$t = 12\sqrt{\frac{0.1 \times 450}{3500}} = 1\tfrac{1}{4} \text{ inches.}$$

This same example was used to find the thickness of the cylinder wall, which was found to be $1\frac{3}{16}$ inches. The usual steam engine practice is to make the cylinder cover the same thickness as the cylinder walls. In the usual type of gas engine the cylinder cover or cylinder head, as it is sometimes called, is very irregular in shape on account of the water jacket and valve chambers. Hence the metal may be made thinner as the section of the cover is more in the nature of a girder. This is shown in Figs. 195, 202 and 203.

Frames. — The material ordinarily used for frames of stationary engines is cast iron although occasionally steel is used. For automobile motors aluminum is frequently used. The allowable fibre stress is low in consideration of the importance of this machine part and because only slight deflections are permissible.

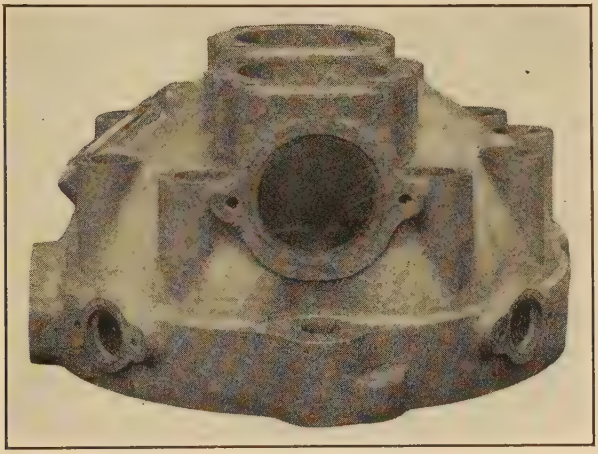


Fig. 203.— Cylinder Cover for a Diesel Engine.

The body of vertical engines is usually called the frame, that of horizontal engines the bed, although the use of these terms is not at all strictly defined. Both of these forms have been fairly well standardized and these standards are closely adhered to by different manufacturers unless necessity demands other construction.

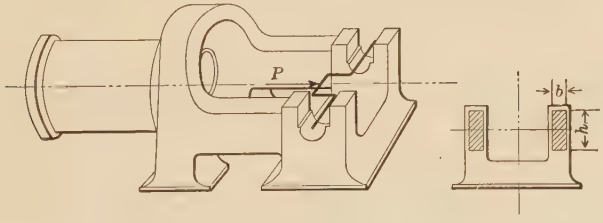


Fig. 204.

The simplest type of frame for the single-acting horizontal engine is shown in Fig. 204. In this type the neutral axis of the two beams connecting the sides of the frame to the bearings and cylinder is in the same horizontal plane as the center of the shaft.

Then these two beams are in direct tension and the stress in them is

$$K_t = \frac{P}{2\,a},$$

where P = total load on piston due to explosion pressure and a is the area of one of the side-beams = bh square inches. The stress K_t should not exceed 1500 pounds per square inch.

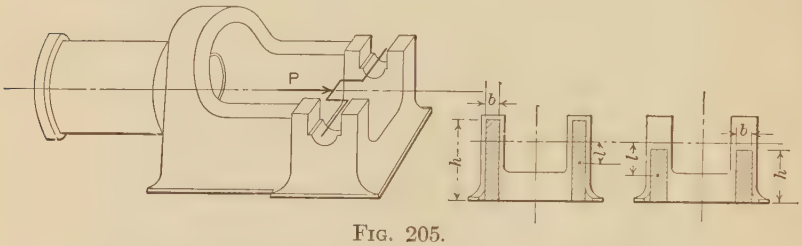


Fig. 205.

The frame illustrated in Fig. 205 is of a type common in small, single-acting engines. The force of the explosion P puts the two side members in tension and bending. The stress in tension will be

$$K_t = \frac{P}{2\,a},$$

as before. In addition there will be a stress due to the bending moment

$$M = Pl,$$

where l is the distance from the center-line of the engine to the center of gravity of the sides of the frame. The moment of inertia of the frame is $\frac{2\,bh^3}{12}$ and the section modulus $\frac{bh^2}{3}$. The bending stress is then

$$K_b = \frac{3\,Pl}{bh^2}.$$

The total stress in the frame will be the sum of the tension and bending stresses, or

$$K_{\text{total}} = K_t + K_b.$$

In medium-size single-acting engines the frame usually has a horizontal member as shown in Fig. 201. In this section the distance l from the center-line of the engine to the center of gravity of the section may be found after x is found.

$$x = \frac{2\,bh \times \frac{h}{2} + fc \times g}{2\,bh + fc}.$$

The moment of inertia of the section is then

$$I = \frac{2\,bh^3}{12} + 2\,bh \times e_1^2 + \frac{cf^3}{12} + cf \times e_2^2.$$

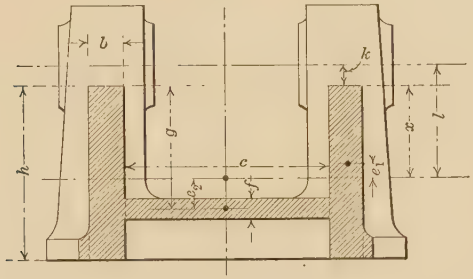


Fig. 206.

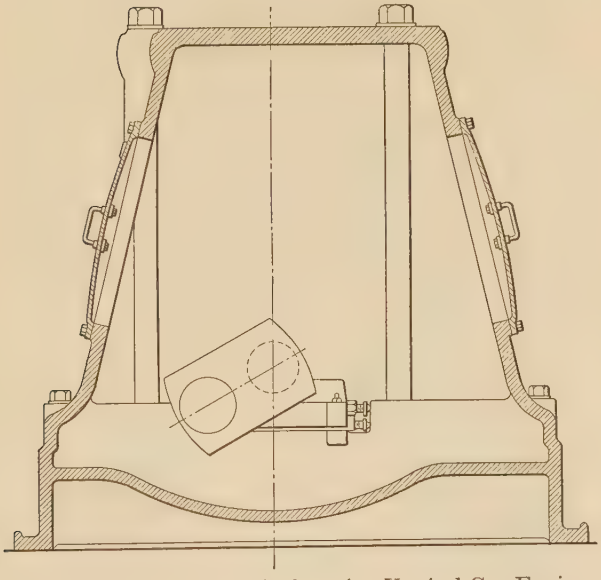


Fig. 207. — Frame for Single-acting Vertical Gas Engine.

The section modulus is $\frac{I}{x}$, and $l = k + x$. The bending stress in the top fibre is then

$$K_b = \frac{Pl}{\frac{I}{x}}.$$

The direct tension is

$$K_t = \frac{P}{2\,bh + fc}.$$

The total stress is

$$K_{\text{total}} = K_b + K_t.$$

It will not be necessary to find the stress in the bottom fibres as they are in compression, due to bending, and tension, due to the direct load P. The net stress will be the difference of these two and will be less than the total stress on the top fibres.

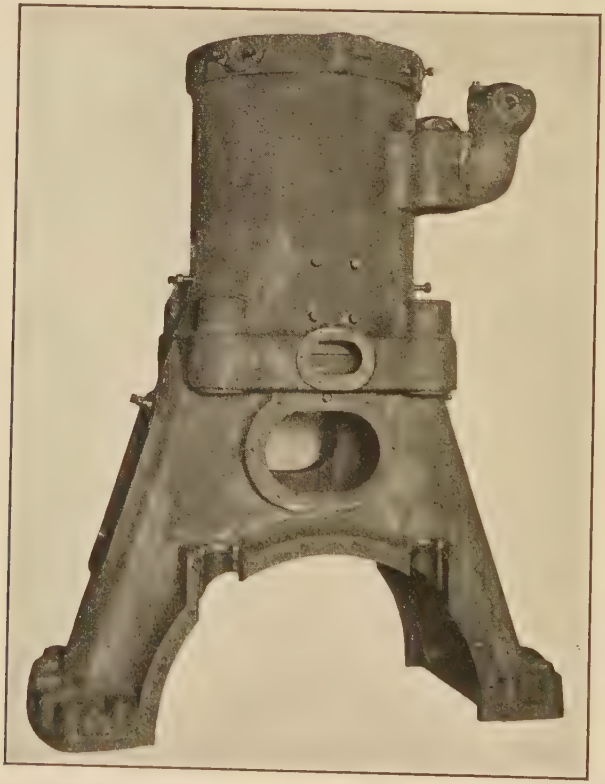


Fig. 208. — Frame and Cylinder for Single-acting Vertical Diesel Engine.

A frame for a vertical multicylinder engine is shown in Fig. 207. In this particular case long steel bolts are employed to take the tension due to the piston pressure. These bolts extend from the bottom of the frame to the plate on which the cylinders are bolted.

The section shown in the illustration was taken between two cylinders.

In Fig. 208 is illustrated a frame for a vertical Diesel engine with the cylinder jacket cast with the frame. The bosses for the tension bolts are shown at the corners. This frame is arranged to be bolted on to a cast-iron sub-base.

Large gas engines usually have a frame which is entirely separate from the cylinder. The one illustrated in Fig. 209 is made by the Snow Steam Pump Works for their tandem engines. The crosshead guides are bored in the frame but the top guide may be removed to facilitate removal of the crosshead. This frame is for a side-crank engine. It will be noticed that the side of the frame opposite the bearing is made high to bring the center of gravity of the frame up near the center-line, thus keeping the bending stress low. Another method of keeping down the bending stress is to make the side members of the frame box-sections instead of solid webs.

It is impossible to use the common method applied to beams in designing engine frames. Instead of assuming a working stress and computing the dimensions it is necessary to assume the dimensions and compute the stress. This cut-and-try method will not prove successful on first trial; very often three or four trials are necessary to find a section that will be both safe and economical.

The bottoms of all frames should have flanges wide enough so that the unit bearing pressure due to weight of machine and whatever inertia forces there may be should not exceed the following figures for the various materials employed:

	Pounds per square inch.
Granite blocks	110–140
Sandstone	70– 80
Hard-pressed brick in cement	55– 70
Concrete	50– 65
Hardwood timber	55– 70
Pine and hemlock timber	30– 40

The attachment of the cylinder to the frame requires careful consideration on the part of the designer. The frame should be bored to receive a turned portion of the cylinder in order to have the two line up perfectly. In Fig. 210 is illustrated a cylinder

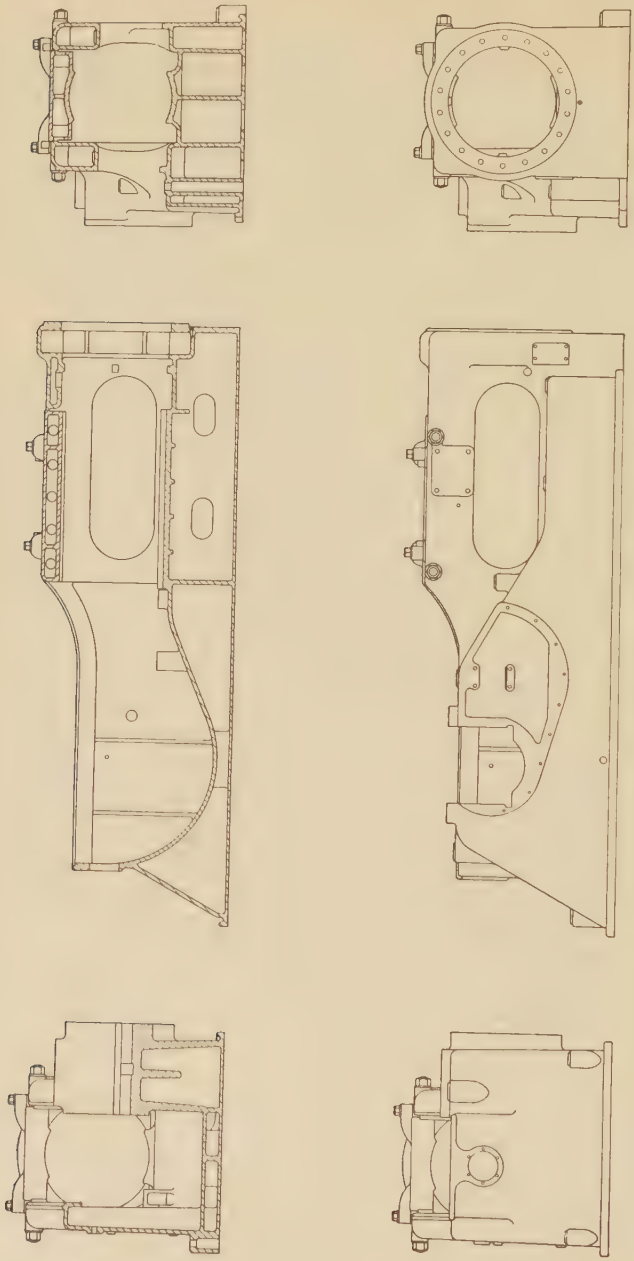


FIG. 209. — Frame for Large Double-acting Gas Engine.

for a two-cycle oil engine. The shoulder *A* was machined to fit into the frame. This shoulder was turned on the boring mill at the same time that the inside of the cylinder was bored.

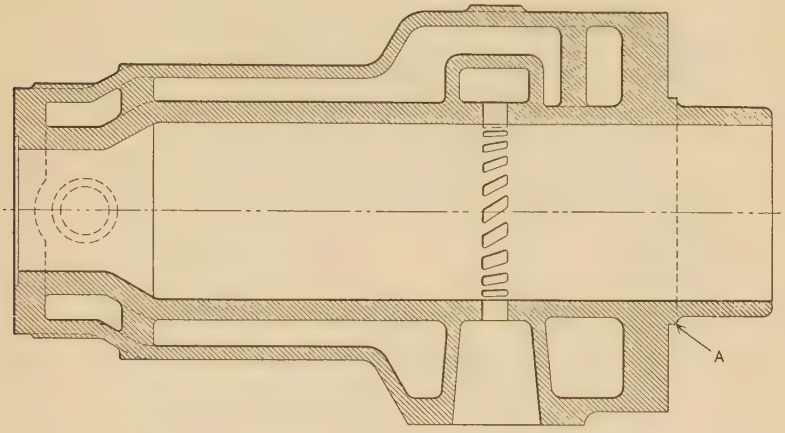


FIG. 210.

Figs. 211, 212 and 213 show details of **cylinder** and **frame connections.** The cylinder in Fig. 211 is cast integral with the jacket and studs are used for connection to the frame. Fig. 212

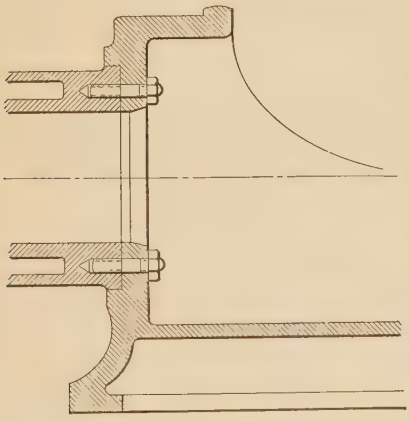


FIG. 211.

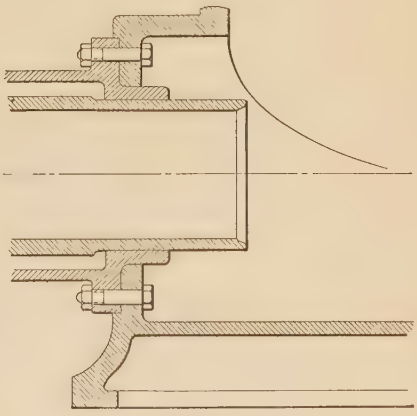


FIG. 212.

illustrates a cylinder with a separate water jacket. The water jacket is bolted to the frame by means of through bolts. The method of casting the jacket with the frame is illustrated in

Fig. 213. The construction used in Figs. 212 and 213 necessitates a water-tight joint where the cylinder extends through the frame. Figs. 214 and 215 illustrate two types of stuffing boxes used at this point. Any kind of soft packing may be used with the one shown in Fig. 214 but a rubber ring should be used with the one in Fig. 215.

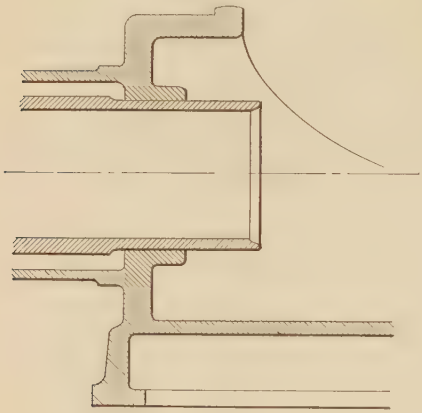


Fig. 213.

The **studs** or **bolts** used in fastening the cylinder head to the cylinder should be made of a good grade of soft steel or wrought iron. The diameter of the studs may be found by equating the total load on the head due to the explosion pressure to the area of all the studs at the root of the thread multiplied by the allowable fibre stress. Thus

$$\frac{\pi d^2}{4} \times p = \left(\frac{n\pi {d_1}^2}{4}\right) K_t \tag{1}$$

and

$$d_1 = d\sqrt{\frac{p}{nK_t}}, \tag{2}$$

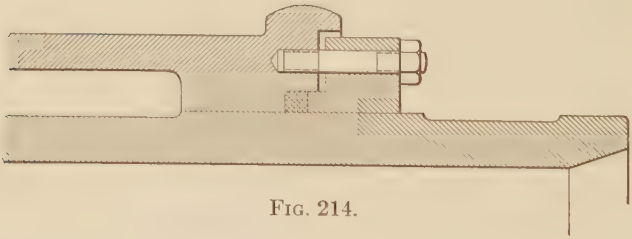


Fig. 214.

where d_1 = diameter of bolt at root of thread,
d = inside diameter of cylinder where head is attached,
p = explosion pressure, pounds per square inch,
n = number of studs,
K_t = allowable fibre stress, from 4000 to 5000 pounds per square inch.

The number of bolts may be about

$$n = 0.25\,d + 4, \tag{3}$$

although many manufacturers used a greater number

$$n = 0.5\,d + 4, \tag{4}$$

where d is the diameter of the cylinder in inches. In order to

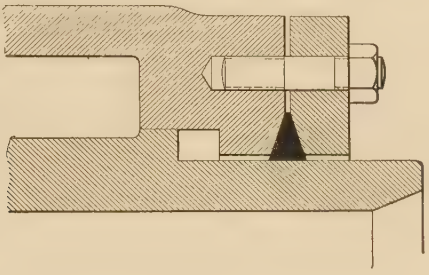


Fig. 215.

prevent the packing from being blown out the maximum distance between studs measured on the pitch circle should be 7 inches.

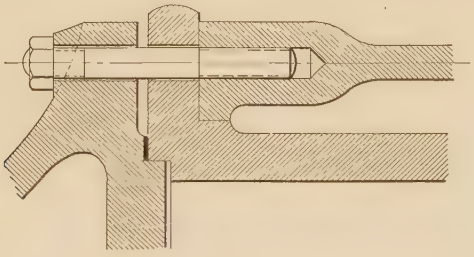


Fig. 216.

The joint between the cylinder cover and cylinder should be made tight by means of packing placed between the cover and

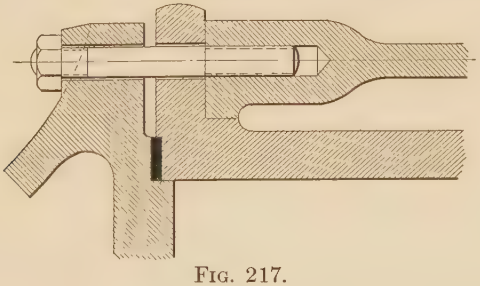


Fig. 217.

cylinder. This packing should be of non-combustible material, either asbestos or soft copper. When asbestos is used it is usually in sheet form while copper may be used in sheet or wire form.

Figs. 216, 217 and 218 illustrate forms of joint that may be used with flat packing. The first is not good as the packing is confined only by the initial tension on the bolts. In the two other cases

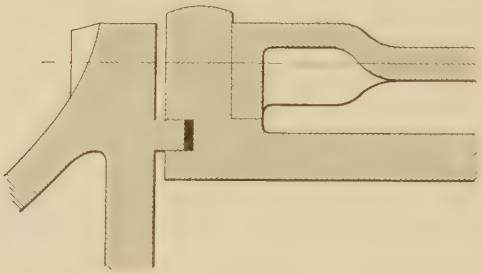


Fig. 218.

the packing is held in a recess between the cylinder and the head. Fig. 200 shows how a round copper ring may be used in place of a flat ring.

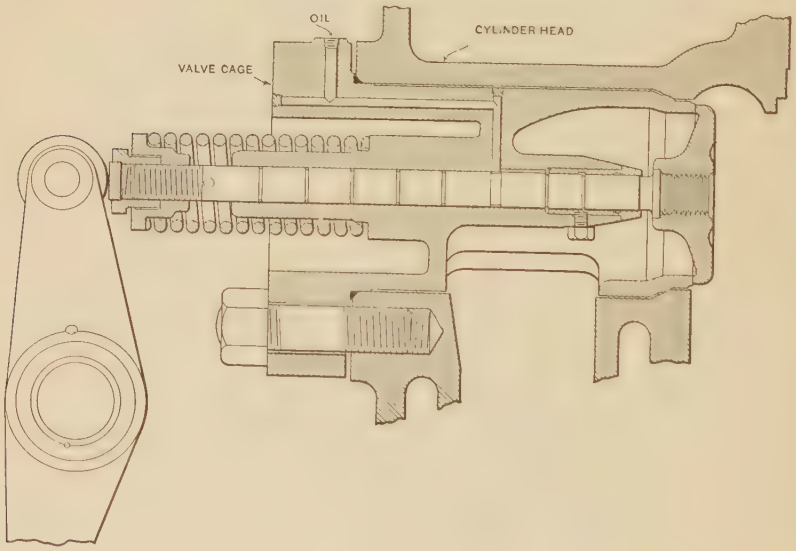


Fig. 219.

Valves. — The material for valve-cages and valve-seats should be hard close-grained cast iron; for valve-discs medium carbon steel or nickel steel; for small automatic valves soft steel may be used. Large valves, especially those water-cooled, are made with cast-iron discs and stems of medium carbon steel.

The **allowable stress** in the valve discs should be very low on account of the high temperatures to which valves are subjected. Thus the discs will be heavy enough to guard against warping and also to allow of regrinding.

The valves of working cylinders of gas engines today are of the poppet type. The inlet-valves may be automatic, but the exhaust valves should be mechanically operated. For all important valves in regular and continuous operation the vertical type should be used. Horizontal valves are much more liable to leakage, sticking and other interruptions of regular service and are especially unreliable when automatic. If the use of the horizontal type is unavoidable the disc of the valve should be made light, the stem guide long and the lift as small as possible. And generally flat seats are to be preferred in horizontal valves rather than the usual conical seat. In Fig. 219 is shown a horizontal valve used as the inlet valve of a horizontal Diesel engine. Fig. 220 illustrates a typical water-cooled exhaust valve used on a large horizontal gas engine. The valve-stem is made up of two concentric tubes. Water enters the bottom end of the inner tube and rises up to the hollow valve disc. Here it circulates in the disc and leaves by means of the annular space surrounding the inner tube. The water connections at the end of the valve-stem must be flexible; they are usually made with heavy rubber hose.

Design of Valves. — Let V be the volume of gas to be passed through the valve per second, v the velocity of the gas in feet per second, h the lift of the valve and d the diameter of the passage, both in feet. Then the free cross-section required through the valve in square feet will be

$$a = \pi\, dh = \frac{V}{v} \text{ square feet.} \tag{1}$$

If the lift of the valve is $h = \frac{d}{4}$, we may write

$$V = 0.7854\, d^2 v, \tag{2}$$

and from this the required diameter of the passage will be

$$d = \sqrt{\frac{V}{0.7854\, v}}. \tag{3}$$

The velocity of gas through the valve should be based on the volume of the piston displacement. This volume per second is

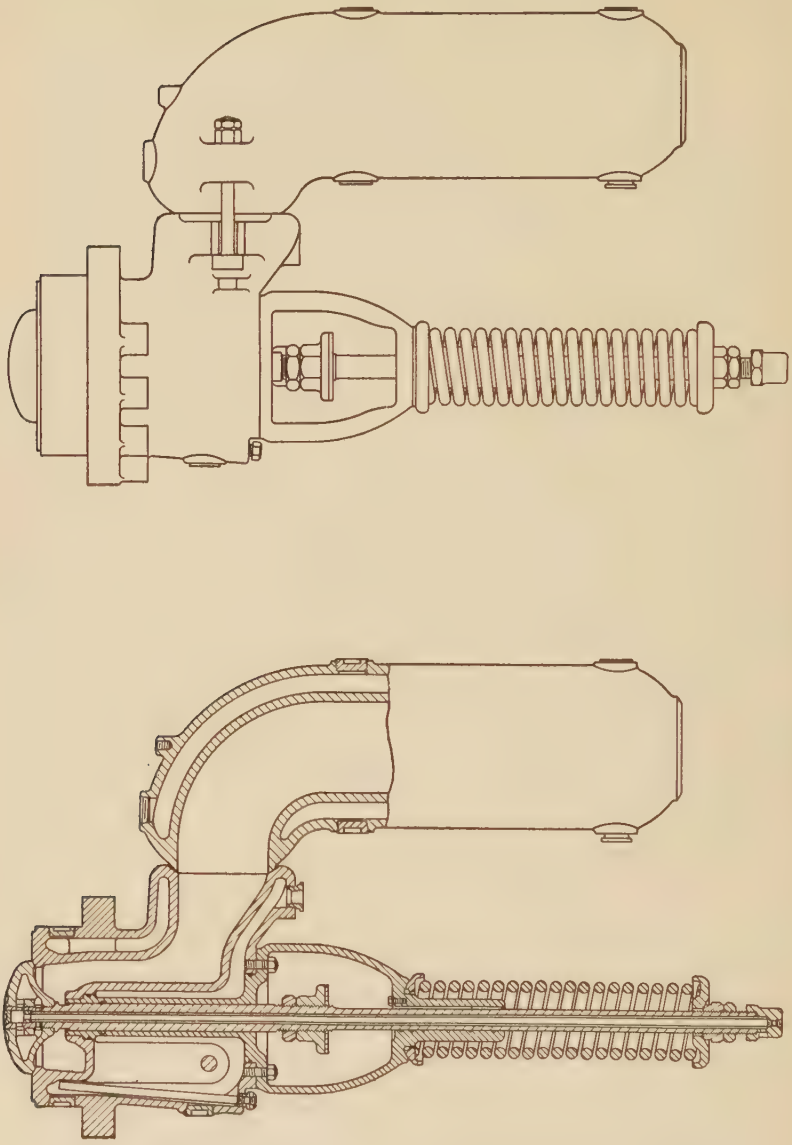


Fig. 220. — Water-cooled Exhaust Valve.

$\frac{2\,ALN}{60}$, where A is the area of the piston in square feet, L the length of the stroke in feet and N the number of revolutions per minute. Then

$$V = \frac{2\,ALN}{60}, \tag{4}$$

and

$$0.7854\,d^2v = \frac{2\,ALN}{60}. \tag{5}$$

Hence

$$d = \sqrt{\frac{2\,ALN}{60 \times 0.7854\,v}}. \tag{6}$$

The mean value of v may be assumed to be 75 feet for small engines and 100 to 120 feet for large engines. These values will answer for both inlet and exhaust valves.

Since the velocity of the piston is not constant the velocity of the gas through the valve will not be constant. Therefore, it is not necessary to have the valve open wide when the piston is moving at a low velocity. In Fig. 221 is shown by the heavy line

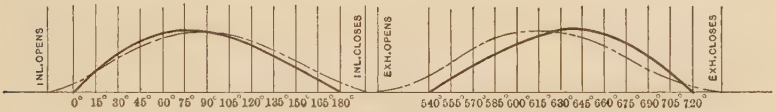


FIG. 221. — Piston Velocity and Valve-lift Curves.

a curve of piston velocity for a gas engine, the abscissæ representing crank angles and the ordinates piston velocities. Drawn on the same base is a curve representing the lift of the valves. The inlet valve usually opens a little before the beginning of the suction stroke and closes after the beginning of the compression stroke. This is to take advantage of any inertia that might exist in the column of gas entering the cylinder. The exhaust valve opens when the crank is from 25 to 40 degrees before the end of the expansion stroke and remains open until after the beginning of the suction stroke. Thus the inlet and exhaust valves are open together for a short time and the inertia of the exhaust gas leaving the cylinder creates a partial vacuum which assists in starting the fresh mixture into the cylinder.

The maximum lift of the valve $\frac{d}{4}$ can be utilized only in the smaller moderate-speed engines. In large engines questions of

valve design and in high-speed engines the proper instant of closing the inlet valve make it necessary to keep the maximum lift much below $\frac{d}{4}$. Automobile engines with automatic inlet valves usually show $h = \frac{d}{10}$ to $\frac{d}{15}$.

If the valve seat is conical instead of flat the actual lift of the valve must be increased in order to make the effective lift equal to h in equation (1). If α is the angle which the seat makes with the vertical the relation between the effective lift and the actual lift will be sin α or

$$\frac{h'}{h} = \sin \alpha, \tag{7}$$

where h = actual and h' = effective lift.

The dimensions of valve-cages and guides depend on general conditions of the particular design. To make gas-tight joints between the cage and cylinder or head to which they are fitted metallic packing rings or surface ground conical at an angle of 60 degrees should be employed. In large cages the metallic packing rings are to be preferred.

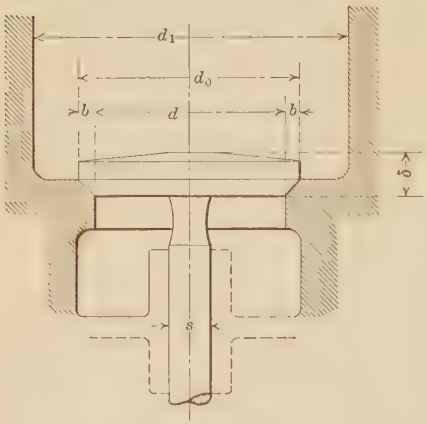


FIG. 222.

As far as the dimensions of the valve disc are concerned, besides d or d_0 only δ, Fig. 222, can be determined by computation. For mechanically operated valves with discs of medium carbon steel we may make

$$\delta = \sqrt{\frac{p_{\max}\,(0.5\,d_0)^2}{6500}} \text{ inches.} \tag{8}$$

This comes from the flat-plate formula for a circular plate supported at the edge and uniformly loaded with a pressure $p_{\max}$, the explosion pressure. The original formula is

$$K = P\left(\frac{r}{t}\right)^2, \tag{9}$$

where K = allowable fibre stress, p = uniform pressure in pounds per square inch, r = radius of plate in inches and t = thickness of plate in inches.

For decreasing the weight of automatic valves K may be taken as high as 11,000 pounds per square inch. In discs which have a diameter greater than 4 inches, the thickness may be decreased to $\frac{2}{3}\,\delta$ at the edges.

The width b of the valve seat may be made approximately

$$b = 0.5\,(d_0 - d) = 0.01\,d + \tfrac{3}{16}\text{ inch.} \qquad (10)$$

The angle of the valve seat is in most cases about 45 degrees. The diameter of the valve stem may be found from the formula

$$s = \tfrac{1}{8}\,d + \tfrac{1}{4}\text{ inch to }\tfrac{1}{8}\,d + \tfrac{7}{16}\text{ inch.} \qquad (11)$$

The stem of the exhaust valve may be made a little greater in diameter in order to decrease the wear and facilitate the conduction of heat to the cooling water. The length of the guide depends upon the design of the valve-cage or housing, and does not depend directly on the size of the valve. If short actuating levers are used on the valves longer guides are necessary than if the reverse were true.

The diameter d_1 of the valve housing must be made large enough so there will be no restriction of the passage around the valve disc. Thus

$$0.7854\,(d_1^2 - d_0^2) > 0.7854\,d^2. \qquad (12)$$

This is satisfied so long as

$$d_1 \geqq 1.6\,d. \qquad (13)$$

It is generally the case, however, that the form of the combustion chamber furnishes more space than is required by equation (13).

Valve Springs. — The valve-springs are generally cylindrical, helical steel springs. The requisite tension on these springs depends mainly on the vacuum occurring in the cylinder. For mechanically-operated valves the effect of this vacuum may be taken equal to a load of from 6 to 8 pounds per square inch of the cross-section of the valve. The springs should have tension enough to prevent the valves opening when the suction is as given above. Thus there will be no gas or air drawn into the cylinder at inopportune times. If the valve-springs should also be employed to bring part of the valve mechanism back into place, the inertia of these parts calls for a corresponding increase in the spring tension. This method of actuating part of the valve-

gear by means of the valve-springs is no longer considered good practice.

In designing the spring care must be taken to avoid overloading the spring when the valve is open. If, with the valve closed, the total load on the spring is W pounds, the load with the valve open will be greater than W because the spring is compressed an amount equal to the valve-lift. The amount that W is increased when the valve is opened depends on the total length of the spring, hence on the number of coils. With the valve wide open there should be from 0.05 to 0.10 inch play between the individual coils.

In discussing the actual design of the spring the following notation will be used:

S = scale of spring = load in pounds required to deflect spring 1 inch,
E = expansion or compression per coil in inches,
d = mean diameter of coil in inches,
W = load in pounds,
δ = diameter of wire or side of square in inches,
D = diameter of wire or side of square in sixteenths of an inch,
G = modulus of elasticity in torsion,
N = number of coils,
K = fibre stress in torsion caused by load W,
C = constant for deflection,
Y = constant for strength.

The relation between load and stress on a helical spring is given by the relation

$$W = \frac{K\pi\delta^3}{\frac{16\,d}{2}} = \frac{\delta^3 \times 2\,K\pi}{d \times 16} = \frac{D^3 Y}{d}. \tag{1}$$

The relation between expansion or compression per coil and the load on the spring is given by the equation

$$E = \frac{64\,W\left(\frac{d}{2}\right)^3}{G\delta^4}. \tag{2}$$

Then
$$W = \frac{E\delta^4}{d^3} \times \frac{8\,G}{64} = \frac{E\,D^4 C}{d^3} \tag{3}$$

and
$$E = \frac{d^3 W}{D^4 C}. \tag{4}$$

From (1) and (3)

$$d = \frac{D^3Y}{W} = \sqrt[3]{\frac{D^4CE}{W}}. \tag{5}$$

Then $$D = \sqrt[3]{\frac{Wd}{Y}} \tag{6}$$

and $$Y = \frac{Wd}{D^3}. \tag{7}$$

The scale of the spring is the load divided by the total deflection in inches. Then

$$S = \frac{W}{EN}. \tag{8}$$

From (4) $$W = \frac{E\,D^4C}{d^3}.$$

Hence $$S = \frac{E\,D^4C}{d^3EN} = \frac{D^4C}{d^3N} \tag{9}$$

and $$N = \frac{D^4C}{Sd^3}. \tag{10}$$

In order to use the equations given above without making tiresome calculations for each spring, values of Y, C, D^3Y and D^4C are given in Tables XVIII and XIX for round and square wire of various sizes.

TABLE XVIII

Size of wire.		Values of Y.			Values of C.
		Up to $\frac{3}{8}$ inch.	$\frac{7}{16}$ to $\frac{3}{4}$ inch.	$\frac{13}{16}$ to $1\frac{1}{4}$ inches.	Up to $1\frac{1}{4}$ inches.
Stress allowed		60,000	50,000	40,000	.....
Round	Steel	5.75	4.80	3.83	22
	Brass	2.50	.....	.....	11
Square	Steel	6.90	5.75	4.60	30
	Brass	3.00	.....	.....	15

TABLE XIX

Valve Spring Constants

Diameter or Side of Square No. A. W. G.	Decimals	D	D^2	D^4	D^3Y				D^4C			
					Round.		Square.		Round.		Square.	
					Steel.	Brass.	Steel.	Brass.	Steel.	Brass.	Steel.	Brass.
$1\frac{1}{4}$	1.25	20	8000	160,000	30,640	....	36,800	....	3,520,000	....	4,800,000	....
$1\frac{3}{16}$	1.188	19	6859	130,300	26,270	....	31,550	....	2,867,000	....	3,909,000	....
$1\frac{1}{8}$	1.125	18	5832	104,980	22,340	....	26,830	....	2,310,000	....	3,150,000	....
$1\frac{1}{16}$	1.063	17	4913	83,520	18,820	....	22,600	....	1,838,000	....	2,506,000	....
1	1.	16	4096	65,540	15,690	....	18,840	....	1,442,000	....	1,966,000	....
$\frac{15}{16}$	0.9375	15	3375	50,630	12,930	....	15,530	....	1,114,000	....	1,519,000	....
$\frac{7}{8}$	0.875	14	2744	38,420	10,510	....	12,620	....	845,200	....	1,152,000	....
$\frac{13}{16}$	0.8125	13	2197	28,560	8,415	....	10,110	....	628,300	....	856,800	....
$\frac{3}{4}$	0.75	12	1728	20,740	8,294	....	9,936	....	456,200	....	622,100	....
$\frac{11}{16}$	0.6875	11	1331	14,640	6,389	....	7,653	....	322,100	....	439,200	....
$\frac{5}{8}$	0.625	10	1000	10,000	4,800	....	5,751	....	220,000	....	300,000	....
$\frac{9}{16}$	0.5625	9	729	6,561	3,500	....	4,192	....	144,300	....	196,800	....
$\frac{1}{2}$	0.5	8	512	4,096	2,458	....	2,944	....	90,110	....	122,900	....
$\frac{7}{16}$	0.4375	7	343	2,401	1,646	....	1,972	....	52,820	....	72,030	....
$\frac{3}{8}$	0.375	6	216	1,296	1,242	540	1,490	648	28,510	14,260	38,880	19,440
No.00	0.3648	5.837	198.9	1,161	1,144	497.2	1,372	596.6	25,540	12,770	34,820	17,410
$\frac{11}{32}$	0.3438	5.5	166.4	915.1	956.7	416.0	1,148	499.1	20,130	10,060	27,450	13,730
No. 0	0.3249	5.198	140.5	730.0	807.6	351.1	969.1	421.4	16,060	8,030	21,900	10,950
$\frac{5}{16}$	0.3125	5.0	125.0	625.0	718.8	312.5	862.5	375.0	13,750	6,875	18,750	9,375
No. 1	0.2893	4.629	99.19	459.1	570.3	248.0	684.4	297.6	10,110	5,050	13,770	6,887
$\frac{9}{32}$	0.2813	4.5	91.13	410.1	524.0	227.8	628.8	273.4	9,020	4,511	12,300	6,150
No. 2	0.2576	4.122	70.04	288.7	402.8	175.1	483.2	210.1	6,350	3,176	8,660	4,330
$\frac{1}{4}$	0.25	4.0	64.0	256.0	368.0	160.0	441.6	192.0	5,632	2,816	7,680	3,840
No. 3	0.2294	3.67	49.43	181.4	284.2	123.6	341.1	148.3	3,990	1,996	5,442	2,720
$\frac{7}{32}$	0.2188	3.5	42.88	150.1	246.5	107.2	295.8	128.61	3,300	1,651	4,502	2,250
No. 4	0.2043	3.269	34.93	114.2	200.9	87.3	241.0	104.8	2,512	1,256	3,426	1,713
$\frac{3}{16}$	0.1875	3.0	27.0	81.0	155.3	67.5	186.3	81.0	1,782	891	2,430	1,215
No. 5	0.1819	2.91	24.64	71.71	141.7	61.6	170.0	73.93	1,578	788.8	2151.3	1,076
No. 6	0.162	2.592	17.41	45.14	100.1	43.5	120.2	52.24	993.1	496.6	1354	677.1
$\frac{5}{32}$	0.1563	2.5	15.63	39.06	89.84	39.1	107.8	46.88	859.4	429.7	1172	586.0
No. 7	0.1443	2.309	12.31	28.43	70.78	30.8	84.94	36.93	625.4	312.7	852.8	426.4
No. 8	0.1285	2.056	8.691	17.87	49.97	21.7	59.97	26.07	393.1	196.6	536.1	268.0
$\frac{1}{8}$	0.125	2.0	8.0	16.0	46.0	20.0	55.2	24.0	352.0	176.0	480.0	240.0
No. 9	0.1144	1.83	6.129	11.22	35.24	15.32	42.29	18.39	246.7	123.4	336.5	168.2
No.10	0.1019	1.63	4.331	7.059	24.9	10.83	29.88	12.99	155.3	77.65	211.8	105.9
$\frac{3}{32}$	0.0938	1.5	3.375	5.063	19.41	8.437	23.29	10.13	111.4	55.69	151.9	75.95
No.11	0.0907	1.451	3.055	4.433	17.57	7.637	21.08	9.165	97.52	48.76	133.0	66.49
No.12	0.0808	1.293	2.162	2.795	12.43	5.404	14.92	6.485	61.50	30.75	83.9	41.95
No.13	0.072	1.152	1.529	1.76	8.79	3.822	10.55	4.586	38.72	19.36	52.8	26.4
No.14	0.641	1.026	1.108	1.108	6.21	2.7	7.452	3.24	24.38	12.19	33.24	16.62
$\frac{1}{16}$	0.0625	1.0	1.0	1.0	5.75	2.5	6.9	3.0	22.0	11.0	30.0	15.0

Example. — Design a spring for an inlet valve which is to open downward. Diameter of valve 6 inches, weight of valve and stem 27 pounds, or 1.00 pound per square inch of valve.

The total load for which the spring is to be designed will be

$$W = \frac{\pi \times 36\,(7 + 1) \times 1.33}{4} = 292 \text{ pounds.}$$

The load on the spring with valve closed will be

$$\frac{292}{1.33} = 220 \text{ pounds.}$$

Thus the increase in load on the spring due to opening the valve is 33 per cent. If we assume the lift of the valve to be one-fifth the diameter or 1.2 inches the scale of the spring will be

$$\frac{292 - 220}{1.2} = 60 \text{ pounds.}$$

The diameter of the stem for this valve would be

$$\tfrac{6}{8} + \tfrac{3}{8} = 1\tfrac{1}{8} \text{ inches.}$$

The mean diameter of the spring d may be made 3 $\times$ the diameter of the stem or in this case $d = 3\frac{3}{8}$ inches. We may use equation (6) to find the diameter of wire necessary. Assuming the wire for the spring to be round steel with a stress of 60,000 pounds per square inch we find the value of Y to be 5.75. Then

$$D = \sqrt[3]{\frac{292 \times 3.375}{5.75}} = 5.56 \text{ sixteenths.}$$

The nearest wire would be $\frac{1}{32}\frac{1}{2}$ inch diameter. Using that value we find $D^4C = 20{,}130$. Then the number of coils will be

$$N = \frac{20{,}130}{60 \times 3.375^3} = 8.75 \text{ or } 9 \text{ free coils.}$$

Since the diameter of wire used, $\frac{1}{32}\frac{1}{2}$ inch, does not quite agree with the calculated diameter, 5.56 sixteenths, we will use equation (7) to find the allowable load.

$$W = \frac{D^3Y}{d} = \frac{956.7}{3.375} = 284 \text{ pounds.}$$

This load is only a trifle less than the calculated load so it will not be necessary to recalculate the spring.

Allowing for a "dead" coil at each end of the spring the total number of coils will be eleven. The length of the spring with the valve open will be $11\ (\frac{1}{32}\frac{1}{2} + \frac{3}{32}\frac{3}{2}) = 4.8$ inches, allowing $\frac{3}{32}\frac{3}{2}$ inch between coils when compressed. The compression of the spring due to full load will be

$$EN = \frac{W}{S} = \frac{292}{60} = 4.87 \text{ inches.}$$

The free length of the spring will then be

$$4.87 + 4.8 = 9.67 \text{ inches.}$$

The specifications of the spring will then be:

Diameter of wire	$\frac{11}{32}$ inch
Outside diameter of coils	$3\frac{23}{32}$ inches
Number of free coils	9
Length of spring extended	$9\frac{11}{16}$ inches

Design of Cams. — Exhaust- and admission-cams for a large producer gas engine are illustrated in Fig. 223. The inlet valve begins to open 9 degrees before head-end center on the exhaust-stroke and remains open until the crank has gone 10 degrees past crank-end dead center on the compression-stroke. The exhaust

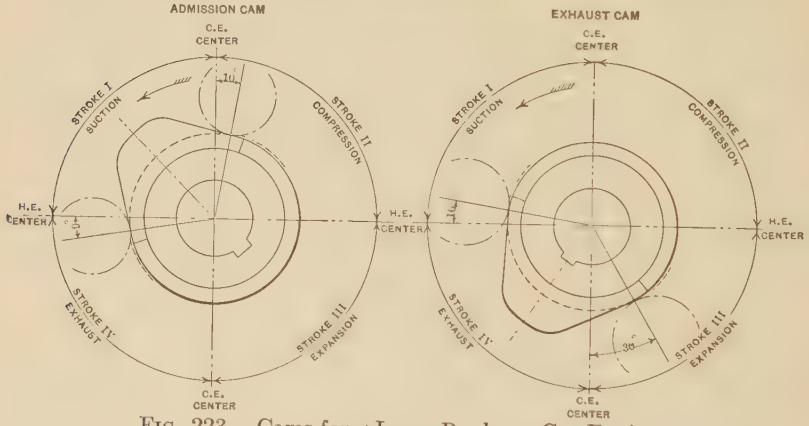


Fig. 223.—Cams for a Large Producer Gas Engine.

valve opens 30 degrees before center on the expansion stroke and remains open until the crank has gone 10 degrees past the head-end center on the suction stroke. Thus the inlet and exhaust valves are open at the same time for 19 degrees of the crank travel.

The cams should be made with hardened surfaces. Sometimes they are made of chilled cast-iron and sometimes of case-hardened machine steel. The rollers are usually made of hardened machine steel. The curve of the cam should be made and the roller set so there will be no jar when the cam and roller engage. There should be a slight clearance between the cam and roller when the valve is closed so that if any particles of grit or dirt fall on the cam it will pass under the roller without opening the valve at inopportune times.

Valve-gears.—To thoroughly cover the design and layout of valve-gears, as the valve operating mechanism is called, would require much more space than is available in this book. There are so many successful types of gears that it would not do to pay attention to only one particular type. Hence a short description of several types is not out of place here.

In Fig. 81 is shown a valve-gear with the inlet and exhaust valves in the cylinder head. The cam-shaft is above the level

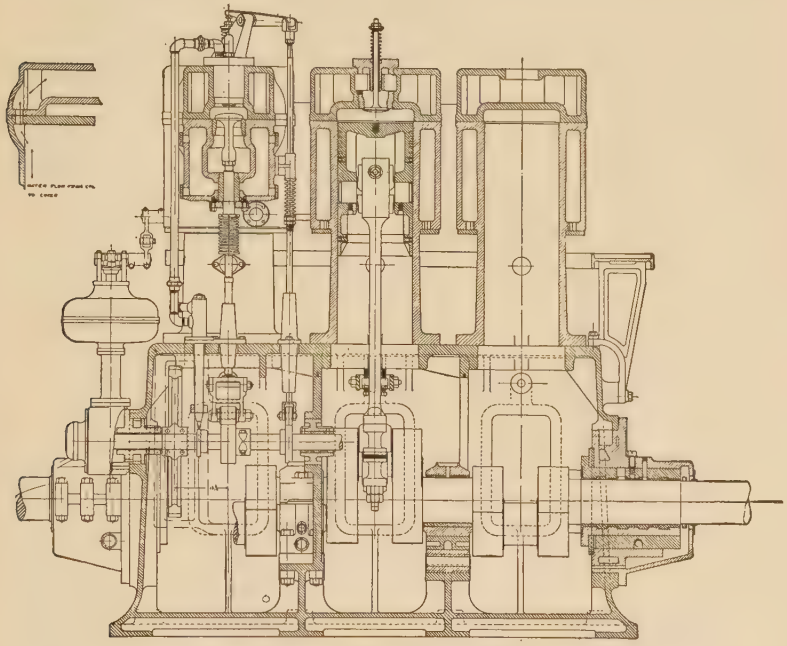


FIG. 224.—Westinghouse Three-cylinder Vertical Gas Engine—Longitudinal Section.

of the cylinder head and is driven by a vertical shaft through bevel gears. The operating arms are short, hence the valve-stems have long guides. Fig. 224 shows a different arrangement of valves in a vertical multicylinder engine. The exhaust valve opens upward and is operated by a long push rod from the cam-shaft which is located in the base of the engine. The inlet valve is in the head and opens downward. This valve is also operated by a long push rod through a short lever placed on top of the cylinder. Fig. 78 illustrates a valve-gear for a single-acting

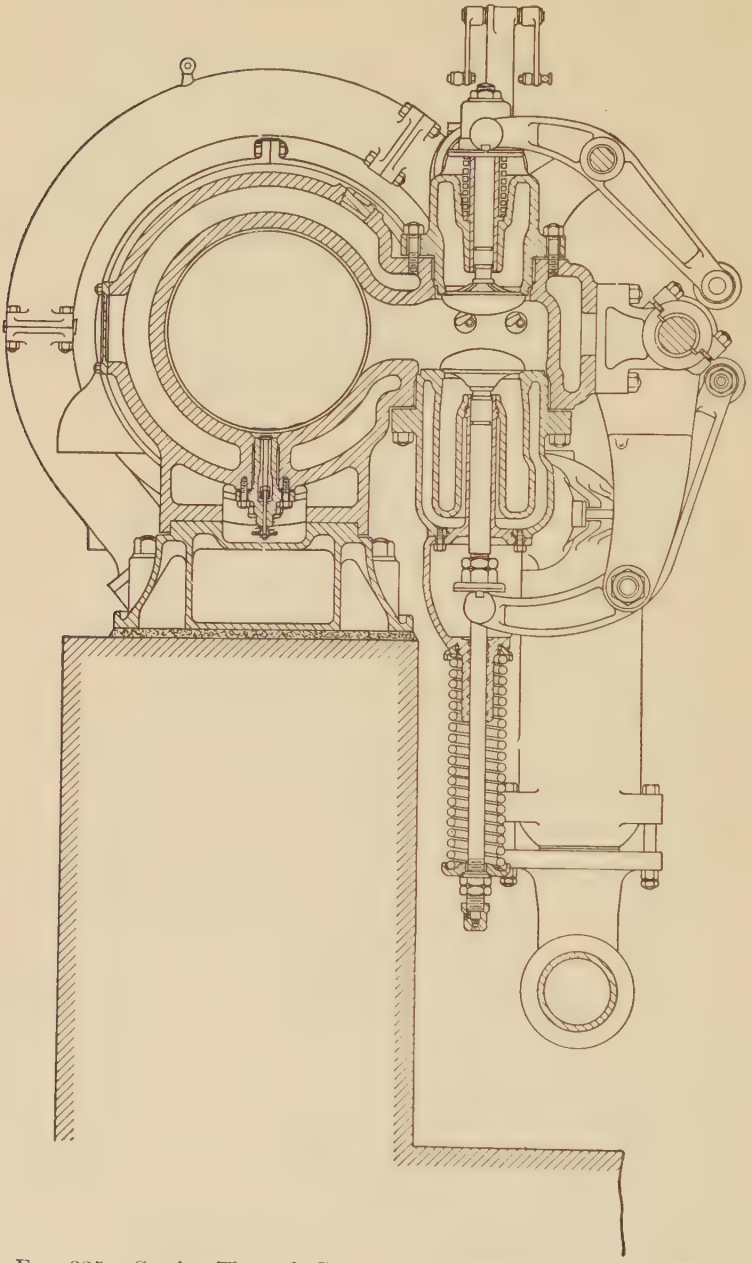


FIG. 225.—Section Through Combustion Chamber. Snow Gas Engine.

horizontal engine of medium size. The exhaust valve is on the bottom of the combustion chamber and opens upward, while the inlet valve is directly over the exhaust valve and opens downward. Both valves are operated through levers from cams on the cam-shaft, which is off at one side and below the center-line of the engine. Each valve has its own cam. The exhaust valve lever is a simple lever pivoted near the middle; the inlet valve lever is also a simple lever, placed up above the level of the top of the cylinder and is horizontal. This lever is actuated by a push rod worked from a bell-crank, one arm of which carries the cam-roller. The push rod carries a knife-edge at its lower end and the position of this knife-edge along the horizontal arm of the bell-crank is determined by the governor. At full load the knife-edge is near the outer end of the bell-crank arm, thus giving the inlet valve *A* full opening. As the load on the engine decreases the knife-edge is drawn in toward the fulcrum of the bell-crank and the opening of the valve is diminished in proportion to the load.

Fig. 225 illustrates the Snow gas engine valve-gear. In this engine the valves are placed in a combustion chamber which is off at one side of the engine center-line. The exhaust valve is below and opens upward; the inlet valve is directly above the exhaust-valve and opens downward. Both valves are driven from the same cam. The cam-shaft is off at one side and on a level with the center-line of the cylinder. The valve levers are I-shaped in section and are made of steel castings. The levers are pivoted so the lift of the valves is less than the "throw" of the cam, the ratio being about 1 to $1\frac{1}{2}$ in both cases.

Fig. 99 illustrates the valve-gear of the Cooper engine. Both valves are driven from the same eccentric, the proper "lift curve" of the valves being secured by rolling levers. This method of operating the valves of large horizontal engines is becoming very popular as it presents some marked advantages over the method of operating by means of cams.

Cam-shaft and Driving Gears.—In small horizontal, single-acting engines the cam-shaft parallel to the cylinder is frequently dispensed with as the valves are arranged so they may be driven by long push rods actuated by cams on short "stub-shafts" which are geared to the main-shaft through spur gears. Cam-shafts are used on all large engines and many small ones but their

design is by means of empirical formulæ rather than by rational methods. The prime requisite for a cam-shaft is rigidity and if it be designed for strength, using the loads which actually come on the shaft, it will not be rigid enough for good operation.

The cam-shaft is usually operated on small engines by means of cylindrical screw or spiral gears, the driver on the crank-shaft being of steel and the driven gear on the cam-shaft of bronze or cast iron. On account of comparatively small surface of contact

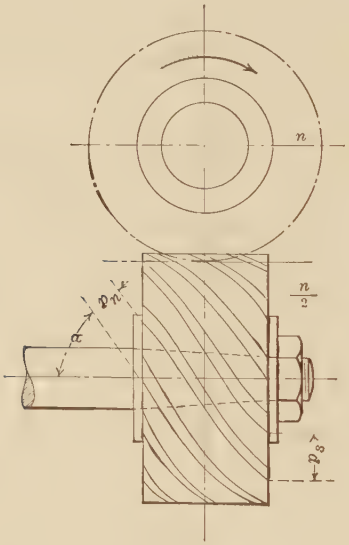


Fig. 226.

and the sliding friction always present in this type of gear, constant lubrication is a necessity. In order to simplify the construction of the bearings it is desirable to have the same diameter of pitch circle for both gears, or at least make the diameter of the driven gear as small as possible, in spite of the fact that the speed ratio is 2:1. For that reason a common tooth angle on the driving gear is 60 degrees and on the follower 30 degrees. In Fig. 226 is shown a sketch of screw gears such as are used for cam-shaft driving. The lateral pitch p_s, if p_n is the normal pitch, will then be

$$p_s = \frac{\pi D}{n} = \frac{pn}{\cos \alpha}. \qquad (1)$$

For the driving gear, when $\alpha = 60$,

$$p_s = \frac{p_n}{0.5} = 2\, p_n. \qquad (2)$$

For the driven gear $\alpha = 30$ and

$$p_s = \frac{p_n}{0.866} = 1.155\, p_n. \qquad (3)$$

The pressure on the teeth of the driven gear causes a thrust on the cam-shaft which must be provided for at one of the bearings, preferably the one nearest the gear. The other bearing should be arranged so the relative lateral motion of the cam-shaft and cylinder, due to heating up of the latter, will not be restricted. The smaller the gear angle on the driven gear the less will be the thrust on the cam-shaft. The width of the teeth or face of the gears is usually from 2 to $2\frac{1}{2}$ p_n. Table XX gives average dimensions of valve-gear parts for small engines.

TABLE XX

AVERAGE DIMENSIONS OF VALVE-GEAR PARTS

B.H.P. of engine.	2	5	10	15	20	30	40	50	60	75	100
p_n	0.65	0.68	0.68	0.71	0.71	0.74	0.74	0.80	0.80	0.87	0.93
d_s	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{3}{4}$	2	2	$2\frac{1}{4}$	$2\frac{1}{2}$
d_r	$1\frac{5}{8}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	4

p_n = normal pitch of gears,
d_s = diameter of cam-shaft,
d_r = diameter of cam-roller, all dimensions in inches.

Cam-shaft gears for large engines require more careful consideration than those for small engines. Many manufacturers do not use spiral gears but prefer to use bevel gears which are sometimes used in connection with spur gears, thus utilizing two pairs of gears to get the 2:1 ratio. In this way each ratio may be made a fractional number with the result that there will be a "hunting tooth" in the system, that is, the load which is applied suddenly as the valve is opened will not always come on the same tooth but will come on all the teeth in turn. With this method the gears wear evenly and last much longer than if a straight 2 : 1 ratio in one pair of gears is used.

Example.—If a cam-shaft is driven from a crank-shaft through a pair of spur gears with 24 teeth on the driving and 25 teeth on the driven gear and a pair of bevel gears with 25 teeth on the driver and 48 teeth on the follower, the total ratio will be

$$\frac{24 \times 25}{25 \times 48} = 2$$

and the wear on the gears will be distributed over all the teeth.

EXERCISES

1. Find the stress in a gas engine cylinder 16 ins. diameter when the explosion pressure is 425 lbs. per sq. in. Thickness of cylinder $1\frac{1}{4}$ ins. (*a*) Due to radial pressure alone. (*b*) Due to radial and longitudinal pressure when Poisson's ratio is 0.25.

2. Find the thickness of cylinder by two methods when the diameter is 14 ins. and pressure is 375 lbs. per sq. in. Find thickness of flange and jacket wall to correspond.

3. Referring to Fig. 199: $D = 18$ ins., $T = 10{,}000$ lbs., $l = 24$ ins., $c = 14$ ins., $c_1 = 18$ ins., $b = 26$ ins., $b_1 = 30$ ins., $t = 1\frac{1}{2}$ ins. Find stress in cylinder due to bending.

4. Referring to Fig. 205: $P = 35{,}000$ lbs., $b = 2\frac{7}{8}$ ins., $h = 32$ ins., $l = 3\frac{1}{2}$ ins., $f = 2$ ins. Find stress in frame.

5. Referring to Fig. 206: $P = 40{,}000$ lbs., $b = 3\frac{1}{2}$ ins., $h = 30$ ins., $c = 26$ ins., $g = 22$ ins., $k = 2$ ins., $f = 2$ ins. Find stress in frame.

6. Compute the number and diameter of studs for a gas engine cylinder 26-in. diameter, pressure 400 lbs. per sq. in.

7. Compute dimensions of valve and stem for a 22×28-in. gas engine running 170 r.p.m. Pressure in cylinder 425 lbs. per sq. in. (*a*) With flat seat. (*b*) With conical seat.

8. Compute spring for an 8-in valve, valve and stem to weigh 45 lbs. Allow 8 lbs. per sq. in. of valve in addition to weight for load on spring. Spring to be round steel with a working stress of 50,000 lbs. per sq. in. Load on spring to increase 25 per cent when valve opens $1\frac{1}{2}$ ins.

CHAPTER XXII

THE CRANK-SHAFT

Side-cranks — Center-cranks — Multiple-cranks

General. — The **crank-shaft** is one of the most vital parts of the engine and hence requires careful consideration in design. No matter what the type of engine, the shaft may be considered as a beam with two or more supports. In addition to the **bending** stress there is always a **shearing** stress present at some time during the stroke caused by the twisting moment on the shaft. Both the bending and the twisting moments are due to the pressure in the cylinder. In addition there are bending moments due to the weight of the flywheel, the weight of the rotor when an engine is direct-connected to an electric generator and the pull of the belts when the engine is belted to a machine of any kind.

It is usually not difficult to determine the magnitudes of the various loads on a crank-shaft but it is very often difficult to determine the distance between supports and the location of the loads in reference to the supports. It is usual to consider the shaft **supported** at the centers of the bearings. The distance between supports then depends on the length of the bearings and the length of the bearings depends on the diameter of the shaft, which necessarily depends on the distance between the supports. This makes it necessary to make an assumption of the lengths of the different bearings before the moments can be calculated. When the diameter is found the lengths of bearings must be recalculated and if the new dimensions are far different from the assumed ones, the diameter of the shaft must be re-calculated. New assumptions must be made and new diameters calculated until the assumption agrees closely with the calculated diameter.

The method of calculation necessarily varies with the type of engine. The center-crank type of shaft requires calculations that are entirely different from those employed with a side-crank

type. Likewise a single-crank shaft requires calculations different from those used with a two-crank or multiple-crank shaft. On account of the numberless combinations of cranks and bearings that we might make it is obviously impossible to cover them all in a general book on gas-engine design. In this book we will cover two or three common types, a side-crank, a center-crank and a multiple-crank shaft. If one is familiar with these three types it will not be difficult to design any other as the theory is the same in all.

There are several ways of finding the bending moments at various points on a beam carrying more than one load. One way is to find the net reactions at the supports due to all the loads. Another method is to find the reactions and bending moments due to the individual loads separately, then combine the bending moments. The author prefers the latter method. It is usually most convenient to find the bending moment for each load by calculation and then draw the individual bending moment diagrams and find the ordinates of the combined bending moment diagram graphically.

The **material** used in crank-shafts is usually open-hearth steel with a tensile strength of 64,000 pounds per square inch and an elongation of at least 20 per cent in 8 inches or mild crucible steel with a tensile strength of 70,000 pounds per square inch and an elongation of 18 or 20 per cent in 8 inches. Of late years nickel steel has been used very freely but it has not been entirely satisfactory as it apparently becomes brittle under repeated straining such as it is subjected to in crank-shafts. The crank-shafts of automobile and other high-speed multicylinder motors are frequently made of nickel steel with chromium added. This steel usually has a tensile strength of from 130,000 to 140,000 and an elastic limit of 110,000 pounds per square inch. The elongation in 2 inches is about 20 per cent and the reduction of area 55 per cent.

The Side-crank Shaft. — This type of shaft is used largely in America for medium-sized and large engines, but does not seem to be in favor in England nor on the continent of Europe. Its chief advantage is that it requires only two bearings no matter whether the engine is a single- or twin-cylinder unit.

The **crank, pin** and **shaft** may be forged in one solid piece or they may be made separately and forced together with hydraulic

pressure. The latter method is usually employed in large engines, the former in medium-sized engines. Fig. 227 shows a typical side-crank forged solid with the shaft. The balance weight, in this case, would have to be bolted to the crank. This method is expensive and is not used on large work. The type shown in

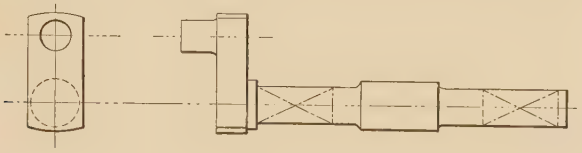


FIG. 227. — Side-crank-shaft Forged in One Piece.

Fig. 228 is used freely. The crank-arm and counterweight are usually made of cast-steel. The crank-pin and the shaft are forced into place under hydraulic pressure. Some manufacturers cast the pin integral with the crank and weight. This method requires a better grade of material in the crank and weight than would be necessary if the pin and crank were made separate.

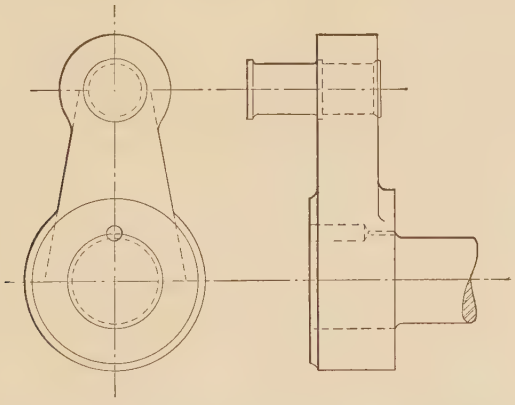


FIG. 228. — Side-crank with Crank Cast Separately.

The crank-shaft should be designed or investigated for at least two positions of the crank, one with the crank-pin on the center at the beginning of the explosion stroke, the other with the crank at the angle where the twisting moment is a maximum. The first case considered here will be with the crank on the center, the shaft to have a flywheel whose weight is W and the flywheel is to be a belt-wheel also, the belt to be horizontal. The distribu-

tion of loads will be as shown in Fig. 229. The size of the crank-pin has been discussed in Chapter XIX. The thickness of the crank-arm e may be assumed to be equal to $\frac{3}{4}\,d_1 + 1$ inch, where d_1 is the diameter of pin, and the distance l equal to $2\frac{1}{2}\,d_1$. Then

$$a = \frac{l_1}{2} + e + \frac{l}{2}.$$

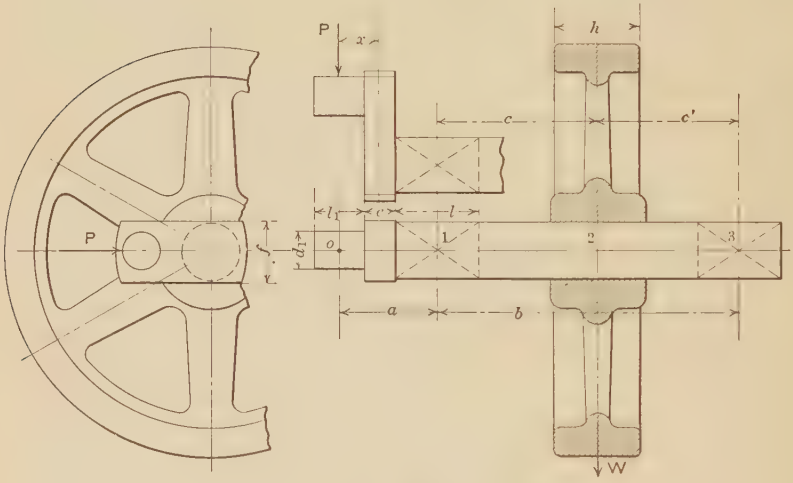


Fig. 229.

The distances c and c' will be determined by the width of the flywheel and space required for gears or eccentrics. The distance c may be found from

$$c = l + \frac{h}{2}.$$

This will leave a distance of $\frac{l}{2}$ inches between the edge of the flywheel and the edge of the bearing for gears. The distance c' may be made less than c, but in many cases is made equal to c.

The diameters at the center of bearing 1 and at the center of the flywheel at 2 should be determined. All loads will be assumed to be in pounds and all dimensions in inches. Using first the load P, the explosion pressure multiplied by the area of the piston, we find the bending moment at 1 to be

$$M_b = Pa \text{ inch-pounds}, \qquad (1)$$

in the horizontal plane if the engine is horizontal. The bending moment at 2 in the horizontal plane due to the belt pull will be

$$M_{2h} = \frac{(T_1 + T_2)\, cc'}{c + c'} \text{ inch-pounds.} \qquad (2)$$

The bending moment at 2 in the vertical plane will be

$$M_{2v} = \frac{Wcc'}{c + c'} \text{ inch-pounds.} \qquad (3)$$

The bending moments found in (1), (2) and (3) should now be laid off on a base representing the distance $a + b$. This horizontal scale may be made anything that is convenient, usually $1\frac{1}{2}$ or 3 inches = 1 foot. The vertical scale, inch-pounds per inch, should be assumed such as to give a maximum height of diagram of about 3 inches. The bending moment diagram of the shaft shown in Fig. 229 is given in Fig. 230. As a matter of convenience the

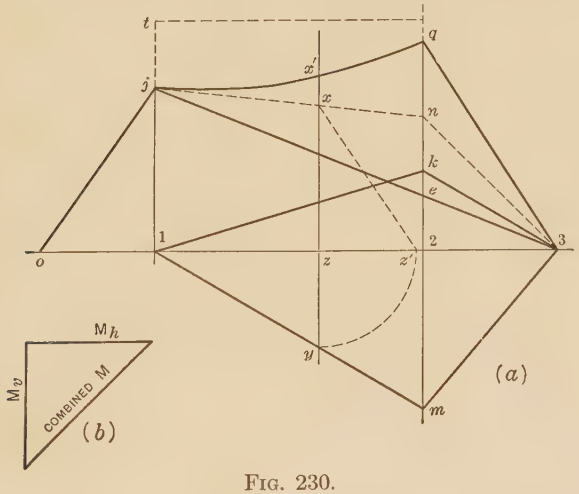


FIG. 230.

vertical moments are laid off below and the horizontal moments above the line. The diagram $1\,m\,3$ is for the flywheel, $2\,m$ being laid off equal to the moment given by equation (3). Diagram $1\,k\,3$ is for the belt pulls, $2\,k$ being equal to the moment given by equation (2) and $oj\,3$ for the piston thrust, $1\,j$ being equal to the moment given by equation (1). Since $oj\,3$ and $1\,k\,3$ are horizontal and tend to deflect the shaft in the same direction, they are added. To do this it is only necessary to lay off kn equal to

$2\,e$ and draw in the dotted diagram $jn\,3$. The ordinates of $ojn\,3$ then represent the total horizontal moments on the shaft at the various points along the axis of the shaft.

To find the diagram of total bending moments it will be necessary to combine $1\,m\,3$ with $ojn\,3$. The most convenient way to do this is to lay off the vertical moment for a given point on the shaft and at right angles to it lay off the corresponding maximum horizontal moment. The closing side of the triangle will then give the magnitude and direction of the combined moment. This is shown at *b*, Fig. 230. Since the direction of the combined moment is of no importance in this particular problem, the combination of the vertical and horizontal moments may be performed on the diagram at *a*, Fig. 230. The method is this: at any point along the base draw an ordinate such as xy, cutting the base at z. With compasses drawn in the arc yz' as shown, xz' is the combined moment and may be laid off on the ordinate xy from z, giving us the point x'. Performing this operation for several points we get the diagram $ojq\,3$, **the ordinates of which represent the total combined bending moments for all points along the shaft.**

In the case illustrated above the maximum moment is $2\,q$ under the flywheel. Using this moment we find the diameter of the shaft to be

$$d = \sqrt[3]{\frac{32\,M_{2q}}{\pi K_t}} \text{ inches.} \tag{4}$$

The condition of loading on this shaft is very severe; the stress on the extreme fibre varies from a maximum tension to a maximum compression and back to a maximum tension with each revolution of the shaft. This rapid reversal of stress necessitates a low fibre stress of 6000 pounds per square inch. The deflection of the shaft also influences the diameter as will be shown later in this chapter. A higher fibre stress would result in a shaft in which the deflection would be extreme.

Stress in the Crank-arm. — The bending moment on the crank-arm is Px, where $x = \frac{l_1}{2} + \frac{e}{2}$. The resisting moment of the arm is $\frac{K_t f e^2}{6}$. The stress will then be

$$K_t = \frac{6\,Px}{fe^2} \text{ pounds per square inch.} \tag{5}$$

The value of f may be assumed to be from $\frac{1}{2}$ to 1 inch greater than the shaft diameter in the bearing. In addition there is a direct compression stress in the crank-arm which is

$$K_c = \frac{P}{fe} \text{ pounds per square inch.} \tag{6}$$

The total stress on the fibres of the side of the crank-arm near the crank-pin will be

$$K_{\text{total}} = K_t + K_c. \tag{7}$$

This total stress should not exceed 10,000 pounds per square inch.

Crank in Position of Maximum Twisting Moment. — After the calculations are made with the crank on the dead center as in the preceding article, the crank angle θ where the maximum twisting moment occurs should be found. This may be done graphically by taking the pressure from the indicator card for different crank angles and finding P_t, Fig. 231. The crank angle which gives the largest value of P_t should be used for θ, Fig. 231.

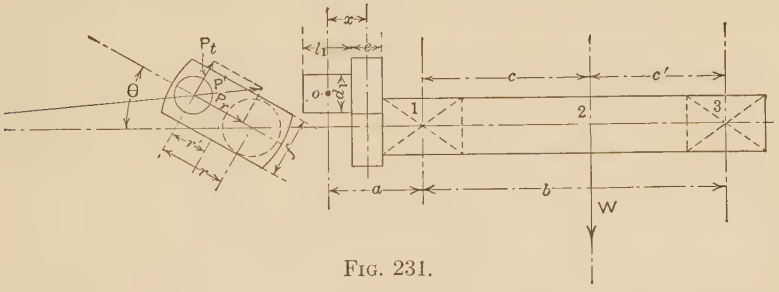


FIG. 231.

If it were not for the fact that the inertia of the reciprocating parts should be neglected in finding P_t maximum we might take the value directly from the rotative effort diagram. During the time in which the engine is starting the inertia forces are small enough to be neglected but the pressure is a maximum during that time. Hence we use the pressure from the card. The angle θ at which P_t is a maximum will probably be between 30 and 40 degrees. After that angle is found a sketch similar to that in Fig. 231 should be drawn and the forces P_r and P_t, radial and tangential components of P' respectively, should be put on, bearing in mind that P' is taken from the indicator card at the piston position corresponding to the crank-angle θ, and that $P' = \frac{p\pi D^2}{4} \sec \alpha$ where p is the pressure per square inch on the piston at crank-angle θ,

d is the diameter of the piston in inches, and α is the angle which the connecting-rod makes with the center-line of the engine.

When a value of $5\frac{1}{2}$ or 6 for the ratio of the length of the connecting-rod to the crank is used, as in most stationary engines, the force P' will be so nearly horizontal that we may call it so without appreciable error. Then the bending moments may be found and combined in precisely the same manner as in Fig. 230, using P' as the load. In addition there will be a twisting moment $P_t r$ on the shaft extending from bearing 1 to the center of the flywheel as shown by the dotted horizontal line in Fig. 230. This constant twisting moment should be combined with the maximum bending moment $2\,q$ to find the equivalent moment.

There are several ways of finding the equivalent bending moment when there are stresses at right angles to each other, as is the case when a shaft is under bending and torsion at the same time. The torsion causes shear on a plane normal to the axis of the shaft and the bending causes tension and compression on a plane parallel to the axis of the shaft. A general condition is illustrated in Fig. 232 which shows direct stresses on the planes 1–2, 4–3, 2–3 and 1–4 of an elementary cube with shear on the planes as shown at right angles to the direct stresses K_t and K_t'. There will be some plane as ab making an angle α with 2–3 on which the stress will be entirely normal. This normal stress is called the **principal stress** and is the greatest stress to which the material is subjected. It can be shown that

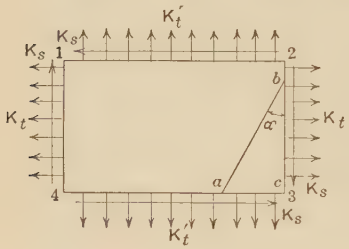


Fig. 232.

$$\tan 2\alpha = \frac{2\,K_s}{K_t - K_t'}, \tag{1}$$

from which two values of α differing by a right angle may be found. It can also be shown that

$$K_{t_{\max}} = \tfrac{1}{2}\,(K_t + K_t') \pm \sqrt{\tfrac{1}{4}\,(K_t - K_t')^2 + K_s^2}. \tag{2}$$

These two values of $K_{t_{\max}}$ are the values of the normal stress intensities on the two principal planes. The larger value (where the upper sign is taken) will be the stress intensity on such a plane as a–b, Fig. 232, and will be the same sign as K_t and K_t'.

The planes on which there are maximum shear stresses are inclined at 45 degrees to the principal planes found and the maximum intensity of shear is

$$K_{s_{\max}} = \sqrt{\tfrac{1}{4}(K_t - K_t')^2 + K_s^2}. \tag{3}$$

The condition existing in the engine shaft is such that K_t' is zero. Then

$$K_{t_{\max}} = \tfrac{1}{2} K_t + \sqrt{\tfrac{1}{4} K_t^2 + K_s^2}. \tag{4}$$

Since $$K_{t_{\max}} = \frac{M_e}{Z},$$

$$K_t = \frac{M}{Z},$$

and $$K_s = \frac{M_t}{Z_p}.$$

We may write

$$\frac{M_e}{Z} = \frac{1}{2}\frac{M}{Z} + \sqrt{\frac{1}{4}\left(\frac{M}{Z}\right)^2 + \left(\frac{M_t}{Z_p}\right)^2}, \tag{5}$$

where M_e is the **equivalent** bending moment causing the maximum or principal normal stress, M is the bending moment causing the stress K_t, M_t is the twisting moment causing the shearing stress K_s, Z is the section modulus $\frac{I}{d_1}$ of a shaft in bending and Z_p is the section modulus of a shaft in torsion. Since $Z_p = 2\,Z$ we may write

$$\frac{M_e}{Z} = \frac{1}{2}\frac{M}{Z} + \sqrt{\frac{1}{4\,Z^2}(M^2 + M_t^2)}, \tag{6}$$

whence $$M_e = \tfrac{1}{2} M + \tfrac{1}{2}\sqrt{M^2 + M_t^2}. \tag{7}$$

Thus, referring to Fig. 230, the bending moment 2–q may be substituted for M and the twisting moment 1–t for M_t in (7) and the result M_e will be a bending moment which would cause the maximum normal stress on the principal plane a–b, Fig. 232. From this equivalent bending moment may be found the diameter of the shaft

$$d = \sqrt[3]{\frac{32\,M_e}{\pi K_t}} \text{ inches}. \tag{8}$$

On the continent of Europe it has been usual to consider the safety of a bar subjected to bending and twisting moments to

depend on the greatest tensile strain or stretch. Then if Poisson's ratio $= \frac{1}{4}$, we have for a circular section

$$M_e = \tfrac{3}{8} M + \tfrac{5}{8} \sqrt{M^2 + M_t^2}. \tag{9}$$

Experiments by Mr. J. J. Guest on ductile materials subjected to combined tension and shear appear to show that failure begins when the shearing stress attains a definite value. While the maximum tension at which yielding occurred in the specimens used by Mr. Guest varied considerably the maximum shearing stress in the different tests was nearly constant. Using equation (3) remembering that K_t' is zero we get

$$K_{s_{\max}} = \sqrt{\tfrac{1}{4} K_t^2 + K_s^2}, \tag{10}$$

and

$$\frac{M_{T_e}}{Z_p} = \sqrt{\frac{1}{4}\left(\frac{M}{Z}\right)^2 + \left(\frac{M_t}{Z_p}\right)^2}. \tag{11}$$

But $Z_p = 2\,Z$. Then

$$\frac{M_{T_e}}{Z_p} = \frac{1}{2\,Z} \sqrt{M^2 + M_t^2}, \tag{12}$$

whence

$$M_{T_e} = \sqrt{M^2 + M_t^2}. \tag{13}$$

This result M_{T_e} is the equivalent **twisting** moment and should be used with the resisting moment in **torsion** in order to find the diameter of a shaft. Then

$$d = \sqrt[3]{\frac{16\,M_{T_e}}{\pi K_s}} \text{ inches}. \tag{14}$$

It should be emphasized here that K_s is the allowable intensity of shear, about 4000 or 5000 pounds per square inch for material used in gas engine shafts.

Stress in the Crank-arm. — The stresses in the crank-arm are more complicated with the shaft in position of maximum turning moment than with the crank on the center. The section where the arm joins the shaft will be the most dangerous as at that point there will be direct compression due to P_r, bending due to P_r and bending due to P_t. The direct compression will be

$$K_c = \frac{P_r}{ef}, \tag{1}$$

the bending stress due to P_t will be

$$K_b = \frac{6\,P_t r'}{ef^2} \tag{2}$$

and that due to P_r will be

$$K_b' = \frac{6\,P_r x}{e^2 f}. \tag{3}$$

The maximum stress will be at a corner on which both the bending stresses found by (2) and (3) will be compression. To these two must be added the direct compression or

$$K_{\text{total}} = K_c + K_b + K_b'. \tag{4}$$

In addition to the compressive stress there is a torsional stress due to the twisting moment $P_t x$. The polar moment of inertia of a rectangle is the sum of the moments of inertia about the two principal axes at right angles. Thus, for the crank under consideration, the polar moment would be

$$I_p = \frac{ef^3}{12} + \frac{e^3 f}{12} = \frac{ef}{12}(e^2 + f^2). \tag{5}$$

Theoretically the distance from the neutral axis to the remote fibre is

$$d_1 = \sqrt{\left(\frac{e}{2}\right)^2 + \left(\frac{f}{2}\right)^2}, \tag{6}$$

making the section modulus

$$\frac{I_p}{d_1} = \frac{\frac{ef}{12}(e^2 + f^2)}{\frac{1}{2}\sqrt{e^2 + f^2}} = \frac{ef}{6}\sqrt{e^2 + f^2}. \tag{7}$$

Actually the fibre at the corner of the section of a rectangular shaft in torsion is not under the greatest stress but the greatest stress falls on the fibre at the middle of the long side. The section modulus is given by Morley's "Strength of Materials," as

$$\frac{I}{d_1} = \frac{a^2 b}{3 + 1.8\frac{b}{a}}, \tag{8}$$

where a is the short and b the long side.

Unwin gives the section modulus as

$$\frac{I}{d_1} = \frac{2}{9} a^2 b. \tag{9}$$

Using the last value we find the torsional stress in the crank-arm to be

$$K_s = \frac{9\,P_t x}{2 f e^2}. \tag{10}$$

In order to find the maximum combined stress in the arm we may use the equation

$$K_{c_{max}} = \tfrac{1}{2} K + \sqrt{\tfrac{1}{4} K^2 + K_s^2}, \qquad (11)$$

where K = the total stress found by equation (4) and K_s = shearing stress found by equation (10). This maximum stress should not exceed 10,000 pounds per square inch.

The diameter of the shaft in the bearings should be determined by the same method used for finding the diameter under the flywheel. At bearing 1 the bending moments due to flywheel and belt pulls will be zero but there will be bending and torsion due to the piston pressure. If the equivalent bending moment is less at 1 than at 2, Fig. 231, the shaft diameter is sometimes made less to correspond but the diameter under the flywheel at 2 is never made less than in the bearing at 1. Usually the bearing at 3 is made the same diameter as at 1. The reactions due to the piston pressure, belt pulls and flywheel at the bearings should be found and the length of the bearings determined, using an allowable pressure per square inch of projected area. The calculations for this work will be found in the latter part of this chapter.

After the lengths of the bearings have been determined new values of a, b, c, c', e, f and x, Fig. 229, may be computed and the shaft diameters recalculated as before. Since these second values will be more accurate than the first assumptions, the second set of diameters found will, in all probability, be the true diameters. However, the dimensions mentioned above should be calculated once more after the second diameters are found and if there is much change a third set of diameters must be found.

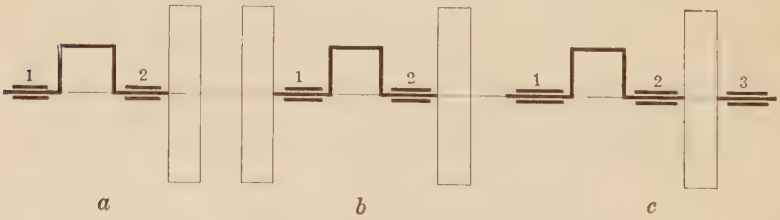


FIG. 233. — Types of Shafts used on Center-crank Engines.

The stresses found in the crank-arm will determine whether any change may be made in e and f. It is obvious that f must be greater than the diameter of the shaft at bearing 1.

The Center-crank Shaft. — There are at least three types of crank-shafts used with single- or tandem-cylinder center-crank engines. These are illustrated in Fig. 233. (*a*) and (*b*) are used only on small engines while (*c*) is used on either small or large engines. In this discussion we will confine ourselves to the type (*c*). The principles of all three are the same and an understanding of one will enable a student to calculate the other two.

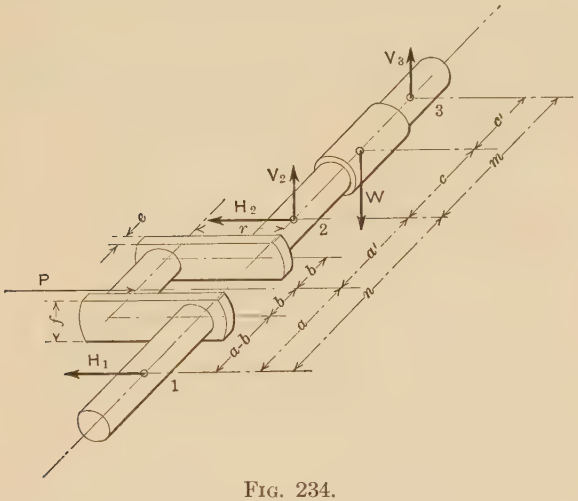


FIG. 234.

Crank on Center. — The shaft of a horizontal engine with its loads and dimensions will appear as in Fig. 234. It is the usual practice to consider that the piston pressure does not have any influence in the bending stress between bearings 2 and 3 and that the weight of the flywheel and the pull of the belt do not have any influence on the bending between bearings 1 and 2.

The Crank-pin. — The bending-moment at the center of the crank-pin is

$$M_b = H_1 a \text{ inch-pounds,} \tag{1}$$

$$H_1 = \frac{Pa'}{n} \text{ pounds.} \tag{2}$$

The shaft is usually made so that a and a' are equal. Then the bending moment at the center of the crank-pin is

$$M_b = \frac{Pn}{4} \text{ inch-pounds.} \tag{3}$$

The length n should be assumed to be 1.8 to 2.2 D, where D is the diameter of the cylinder.

The diameter of the crank-pin will be

$$d = \sqrt[3]{\frac{32\,M_b}{\pi K_t}} \text{ inches.} \qquad (4)$$

The allowable stress should be 10,000 or 12,000 pounds per square inch.

Left-hand Crank-arm. — After the diameter of the crank-pin has been determined as above, the dimensions of the crank-arm may be determined. The width of the arm f may be made $1\frac{1}{8}\,d + \frac{1}{2}$ inch and the thickness $e = 0.65\,d + \frac{1}{4}$ inch. The bending moment on the arm is

$$M_b = H_1\,(a - b) \text{ inch-pounds.} \qquad (1)$$

From this the stress may be found

$$K_b = \frac{6\,M_b}{e^2 f} \text{ pounds per square inch.} \qquad (2)$$

The stress due to direct compression will be

$$K_c = \frac{H_1}{ef}.$$

The total stress will be

$$K_{\text{total}} = K_b + K_c.$$

This stress should not exceed 10,000 pounds per square inch.

Right-hand Crank-arm. — The bending moment on the right-hand crank-arm will be

$$M_b = H_1\,(a + b) - Pb \text{ inch-pounds,} \qquad (1)$$

which is the same as that on the left-hand arm, and since the size of the arm is the same the stress will be the same. This stress is on the face of the broad side of the arm.

Shaft Under Flywheel. — The moments due to the flywheel in the vertical and the belt pull in the horizontal plane may be found as for the side-crank shaft. These two moments should be combined at right angles and the corresponding diameter of the shaft found. Thus if M_w is the bending moment due to the flywheel and M_b that due to the belt the total bending moment will be

$$M_{\text{total}} = \sqrt{M_w^2 + M_b^2} \text{ inch-pounds.} \qquad (1)$$

The diameter of the shaft will be

$$d = \sqrt[3]{\frac{32\,M_{\text{total}}}{\pi K_t}} \text{ inches,} \qquad (2)$$

where K_t is the allowable stress of 6000 pounds per square inch.

Crank at Angle of Maximum Twisting Moment. — Fig. 235 illustrates the center-crank shaft at a position where the twisting moment is a maximum. This position should be found as stated in the discussion of the side-crank shaft. After the total force P' is found for this position, the tangential and horizontal components P_t and P_r should be found graphically.

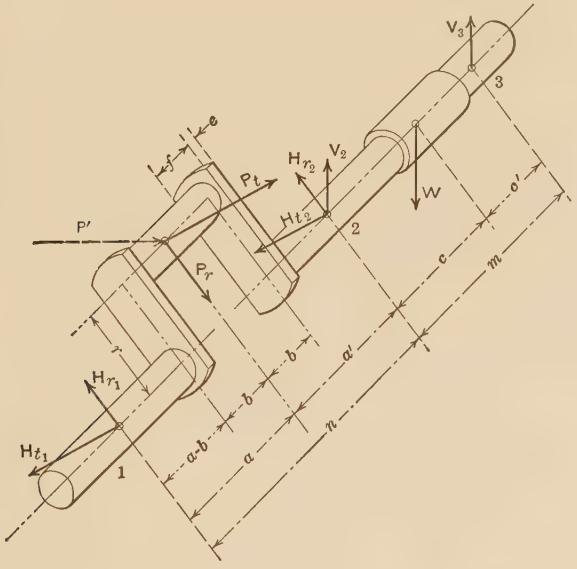


FIG. 235.

Shaft Under Flywheel. — The maximum combined bending moment $M_{b_{\max}}$ due to the weight of the flywheel and the pull of the belt should be found as before. The twisting moment will be

$$M_t = P_t r \text{ inch-pounds.} \qquad (1)$$

The equivalent twisting moment will be

$$M_{t'} = \sqrt{M_t^2 + M_{b_{\max}}^2} \text{ inch-pounds.} \qquad (2)$$

Using this moment we find the diameter of the shaft under the flywheel to be

$$d = \sqrt[3]{\frac{16\,M_{t_e}}{\pi K_s}} \text{ inches,} \tag{3}$$

where K_s is the allowable stress in shear, about 5000 pounds per square inch.

Shaft at Juncture of Right-hand Crank-arm. — The twisting moment here is the same as that given under the flywheel. The bending moment will be

$$M_b = H_1\left(a + b + \frac{e}{2}\right) - P'\left(b + \frac{e}{2}\right) \text{inch-pounds,} \tag{1}$$

$$\text{where } H_1 = \frac{P'}{2}, \text{ the reaction at 1 due to } P'.$$

The twisting and bending moments should be combined to find the equivalent twisting moment using Guest's law as before, and this equivalent moment used to find the diameter using a stress of 5000 or 6000 pounds.

Crank-pin. — The crank-pin is subjected to bending and twisting in the position shown in Fig. 235. The bending moment is due to H_1 and is

$$M_b = H_1 a \text{ inch-pounds.} \tag{1}$$

The twisting moment is due to H_t and is

$$M_t = H_{t_1} r \text{ inch-pounds.} \tag{2}$$

The equivalent twisting moment may be found by combining (1) and (2) according to Guest's law. Then

$$M_{t_e} = \sqrt{M_b^2 + M_t^2} \text{ inch-pounds.} \tag{3}$$

Then the diameter will be

$$d = \sqrt[3]{\frac{16\,M_{t_e}}{\pi K_s}} \text{ inches,} \tag{4}$$

where K_s is the allowable shearing stress in pounds per square inch, about 5000 or 6000.

Left-hand Crank-arm. — The left-hand crank-arm is subjected to bending due to H_{r_1} and H_{t_1}, to twisting due to H_{t_1} and to direct compression due to H_{r_1}.

The bending moment due to H_{r_1} is

$$M_{b_r} = H_{r_1}(a - b) \text{ inch-pounds.} \tag{1}$$

The corresponding stress is

$$K_{b_r} = \frac{6\,M_{b_r}}{e^2 f} \text{ pounds per square inch.} \tag{2}$$

The bending moment due to H_{t_1} is

$$M_{b_t} = H_{t_1} r' \text{ inch-pounds,} \tag{3}$$

at the juncture of the crank-pin and arm, where $r' = r - \frac{1}{2}$ diameter of crank-pin.

The corresponding stress is

$$K_{b_t} = \frac{6\,M_{b_t}}{ef^2} \text{ pounds per square inch.} \tag{4}$$

The stress in direct compression is

$$K_c = \frac{H_{r_1}}{ef} \text{ pounds per square inch.} \tag{5}$$

The total compression stress on the fibres on one corner will be

$$K_{c_{\text{total}}} = K_{b_r} + K_{b_t} + K_c. \tag{6}$$

In addition to the above stresses there is a stress in torsion. The twisting moment is

$$M_t = H_{t_1}(a - b) \text{ inch-pounds.} \tag{7}$$

The corresponding stress is

$$K_s = \frac{9\,M_t}{2fe^2} \text{ pounds per square inch.} \tag{8}$$

The total combined stress on the arm will then be

$$K_{c_{\max}} = \frac{1}{2}K + \sqrt{\frac{K^2}{4} + K_s^2}, \tag{9}$$

where K = the total stress found by equation (6) and K_s = shearing stress found by equation (8). The maximum stress as found by equation (9) should not exceed 10,000 pounds per square inch.

Right-hand Crank-arm. — The crank-arm on the right or towards the flywheel is subjected to bending stresses in two planes normal to each other and also to torsion. The bending moment due to the radial pressure is constant throughout the length of the crank-arm. It is

$$M_{b_r} = H_{r_1}(a + b) - P_r b \text{ inch-pounds.} \tag{1}$$

The stress to correspond is

$$K_{b_r} = \frac{6\,M_{b_r}}{e^2 f} \text{ pounds per square inch.} \tag{2}$$

The bending due to the tangential component is a maximum where the arm joins the shaft, hence the total stress will be a maximum at that point. The bending due to the tangential component is

$$M_{b_t} = P_t r'' + H_{t_1} \times \text{radius of shaft at 2 inch-pounds,} \quad (3)$$

where $r'' = r - \frac{1}{2}$ diameter of shaft at 2.

The stress to correspond will be

$$K_{b_t} = \frac{6\,M_{b_t}}{ef^2} \text{ pounds per square inch.} \quad (4)$$

The stress in direct crushing is

$$K_c = \frac{P_r}{2\,ef} \text{ pounds per square inch.} \quad (5)$$

The twisting moment on this arm is

$$M_t = H_{t_1}(a + b) - P_t b \text{ inch-pounds,} \quad (6)$$

and the corresponding stress is

$$K_s = \frac{9\,M_t}{2\,fe^2} \text{ pounds per square inch.} \quad (7)$$

The total compressive stress should be found by adding the stresses found separately by equations (2), (4) and (5). Thus

$$K_{\text{total}} = K_{b_r} + K_{b_t} + K_c. \quad (8)$$

The maximum stress may now be found by equation (9) of the last article. This maximum stress should not exceed 10,000 pounds per square inch.

After the above calculations have been made the lengths of the bearings should be determined and the new values of a, a', b, c, c', e and f determined. If these new values differ materially from those assumed for the first calculation, new calculations must be made, using the new dimensions.

When the diameters of the shaft at the various points have been determined, the stresses at bearing 2, at the crank-pin and under the flywheel should be found, using the theorem of 3 moments. This should be done with the shaft in position of maximum torque. The loads are then P' at the center of the crank-pin in the horizontal direction and W under the flywheel in a vertical direction. There is also a twisting moment between bearing 2 and the flywheel equal to $P_t r$. In this work it will not be necessary to take into account the belt tensions.

Fig. 236 shows disposition of loads on a shaft to be used for calculation by the theorem of 3 moments. The equation to be used is

$$M_1L' + 2M_2(L' + L'') + M_3L'' = -P'L'^2(K' - K'^3) - WL''^2(2K'' - 3K''^2 + K''^3),$$

where M_1, M_2 and M_3 are the bending moments at bearings 1, 2 and 3 respectively, L' and L'' are the dimensions shown on Fig. 236, K' is the ratio of a to L' and K'' the ratio of c to L''. This

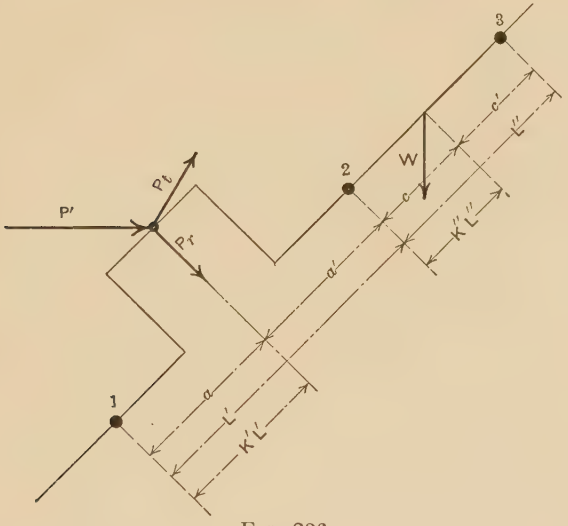


FIG. 236.

equation should be used to find M_2, the bending moment at bearing 2. In solving this equation it is obvious that the bending moments at 1 and 3, M_1 and M_3, are zero. Hence

$$M_{p_2} = -\frac{P'L'^2(K - K^3)}{2(L' + L'')} \text{ inch-pounds}$$

for the piston pressure alone. When using the flywheel weight P' is zero and

$$M_{w_2} = -\frac{WL''^2(2K'' - 3K''^2 + K''^3)}{2(L' + L'')} \text{ inch-pounds.}$$

The reactions should be found for each case so the complete bending moment diagrams may be drawn.

$$R_{p_1} = \frac{M_2}{L'} + P'(1 - K),$$

$$R_{p_3} = \frac{M_2}{L''},$$

$$R_{p_2} = P' - R_{p_1} - R_{p_3},$$

when P' is considered.

$$R_{w_1} = \frac{M_2}{L'},$$

$$R_{w_3} = WK'' - \frac{M''}{L''},$$

$$R_{w_2} = W - R_{w_3} - R_{w_1},$$

when W is considered. In finding these reactions it is necessary to pay particular attention to the signs.

Since the load P' is horizontal and the load W vertical it will be more convenient to find the bending moments for the two separate and plot them with the twisting moment. Then the resultant of the vertical and horizontal bending moments may be found and combined with the combined bending moment to find the equivalent twisting moment. In order to illustrate the calculations an example is given at the end of this chapter.

Multiple-throw Crank-shafts. — The design of multiple-throw crank-shafts may be based on the principles set forth for the single-throw center- and side-crank types. When the horizontal twin-type engine is used for stationary practice the cranks are overhung and set at 90 degrees from each other and there are only two bearings on the shaft. This makes it unnecessary to use the theorem of 3 moments. The moments should be found and the diagrams drawn for at least two positions, one with one crank on the center and the other crank 90 degrees ahead, the other position with one of the cranks at the position of maximum twisting moment.

Vertical multicylinder engines usually do not have overhung cranks but there is a bearing at each end of the shaft. Since the maximum twisting moment for each crank always occurs within the first one-fifth of the expansion stroke, no two maximum twisting moments can occur at the same time in any of the multicylinder combinations now used. For that reason the diameter of the shaft may be calculated on the basis of one maximum twisting moment as before.

Fig. 237 illustrates a crank shaft for a four-cylinder vertical marine engine. The inclined position of the middle crank-arm must be regarded as a makeshift only to which recourse must be had when the cylinders are so close together that there is no room for a bearing between. In general this construction should be avoided because the inclined arms are subject to the action of additional

forces which not only increase the stresses but complicate the method of computation. In this case the cylinders "fire" in the order I, IV, II, III.

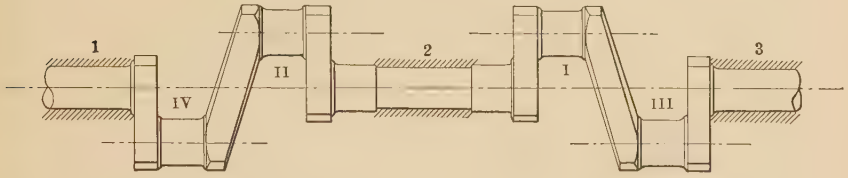


Fig. 237. — Crank-shaft for a Four-cylinder Marine Engine.

The method of procedure in computing a multiple-throw crank-shaft should be as follows:

First, compute dimensions of crank-pins for bearing pressure and strength, find the distances between bearings 1–2 and 2–3 necessary to conform to cylinder dimensions.

Second, assume dimensions of crank-arms.

Third, with shaft in position as shown in Fig. 237, with crank II at the beginning of the explosion stroke, compute reactions at 1 and 2, bending stress in the arm at the left which is toward the load and in the inclined arm at the right.

Fourth, repeat all the calculations under the third item with crank II at the angle of maximum twist and in addition find the twisting in the pin and arms.

Fifth, fix size of bearings to take the load upon them and check stress in them using the theorem of three moments.

If a multicylinder engine has a heavy flywheel the weight should be taken into account if there is no outboard bearing. If there is an outboard bearing the shaft between the end bearing of the engine and the outboard bearing should be designed as a separate beam carrying the flywheel and generator if direct-connected, and belt pulls if belted, in addition to a twisting moment from the engine.

The calculations required above present nothing new except the computation of the stress in the inclined arm. In order to make those clear an example is given here:

Fig. 238 illustrates the forces on the left-hand inclined crank-arm of Fig. 237, crank II on dead center. In this example the shaft will be treated as a simple beam with a load on crank-pin II only. The piston pressure P is 27,000 pounds and the reaction

R is 15,500 pounds. The piston pressure and reaction are resolved parallel and normal to the inclined crank-arm. The components are $P_t = 9200$ pounds, $P_r = 25{,}400$ pounds, $R_t = 5300$

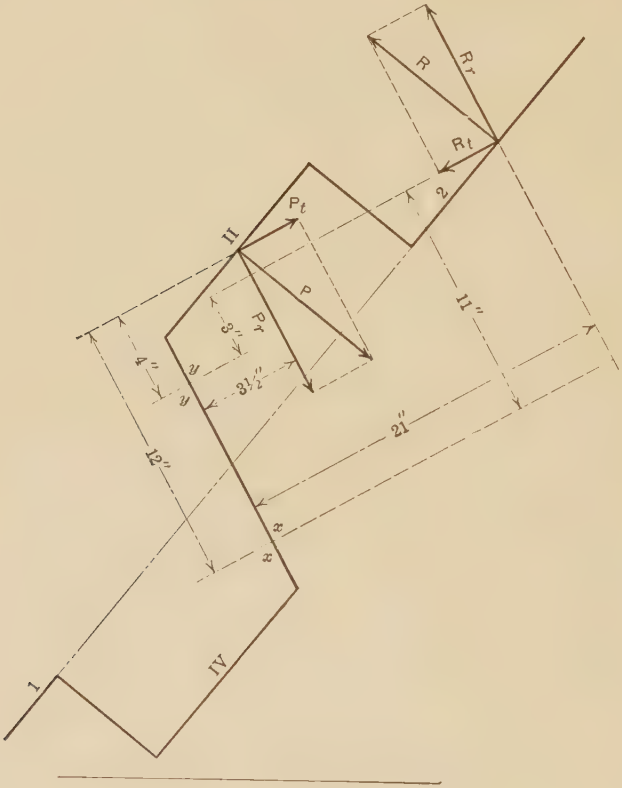


Fig. 238.

pounds and $R_r = 14{,}700$ pounds. The dangerous sections of the arm are at x–x and y–y. The bending moments at x–x are

$$M_{b_1} = 25{,}400 \times 3.5 = 89{,}900 \text{ inch-pounds},$$

and $$M_{b_2} = 14{,}700 \times 21 = 309{,}000 \text{ inch-pounds}.$$

Since these are in opposite directions they must be subtracted, giving the net bending moment due to the parallel components

$$M_{b_r} = 309{,}000 - 89{,}900 = 219{,}100 \text{ inch-pounds}.$$

The bending moments due to the normal components are

$$M_{b_3} = 9200 \times 12 = 110{,}000 \text{ inch-pounds,}$$

and $$M_{b_4} = 5300 \times 11 = 58{,}000 \text{ inch-pounds.}$$

These also are in the opposite direction and must be subtracted.

$$M_{b_t} = 110{,}000 - 58{,}000 = 52{,}000 \text{ inch-pounds.}$$

It is evident from the figure that M_{b_r} and M_{b_t} are opposite in sign and hence the result is the difference. That is

$$M_{b_{net}} = 219{,}100 - 52{,}000 = 167{,}100 \text{ inch-pounds.}$$

The size of the arm is $6\frac{3}{4} \times 3\frac{3}{16}$ inches. The section modulus is then $\frac{1}{6} \times 6.75 \times \overline{3.8125}^2 = 16.3$ inches^3. The stress in the arm due to the radial load will then be

$$K_{b_x} = \frac{167{,}100}{16.3} = 10{,}250 \text{ pounds per square inch.}$$

The stress at the section y–y may be found in the same manner. The bending moments due to P_r and R_r will be the same as at the section x–x, the net moment being

$$M_{b_r} = 219{,}100 \text{ inch-pounds.}$$

The moment due to P_t is

$$M_{b_3} = 9200 \times 4 = 36{,}800 \text{ inch-pounds.}$$

That due to R_t is

$$M_{b_4} = 5300 \times 3.125 = 16{,}600 \text{ inch-pounds.}$$

The difference of these is

$$M_{b_t} = 38{,}600 - 16{,}600 = 22{,}000 \text{ inch-pounds.}$$

The net bending moment is

$$M_{b_{net}} = 219{,}100 - 22{,}000 = 197{,}100 \text{ inch-pounds.}$$

The stress is

$$K_{b_y} = \frac{197{,}100}{16.3} = 12{,}100 \text{ pounds per square inch.}$$

Additional Loads on Shaft. — When the gas engine drives an electric generator the weight of the rotor should be included as a load in the vertical plane when the rotor is between the outboard-bearing and the engine frame. In some instances the generator is driven by means of a flexible coupling and therefore does not cause any bending stress in the engine shaft. If such types of engines as are shown in Fig. 233, (*a*) and (*b*), were direct-connected to generators, the generator rotor would be supported on two in-

dependent bearings and driven by a flexible coupling. This would throw no additional load on the engine shaft but would actually relieve the shaft of one of its loads, the belt tensions. If the type shown at (*c*) is direct-connected to a generator then the weight of the rotor would come on the right-hand main-bearing and the outboard-bearing, causing additional stresses in the shaft.

Still another load that must be accounted for is the magnetic pull on the rotor due to the displacement of the shaft from the center-line of the stator of the generator. This displacement is due to two causes, the deflection of the shaft caused by the loads on it and the displacement due to wear in the bearings. After the total displacement is found the magnetic pull may be found after the type of generator is known. This pull must be treated as additional weight at the center-line of the rotor.

Deflection of the Shaft. — If W' is the sum of the loads concentrated at the center of the flywheel the deflection of the shaft will be

$$\frac{W'c^2c'^2}{3\,IEm}\text{ inches},$$

where c, c' and m are dimensions given in Fig. 234, I is the plane moment of inertia of the shaft, E is the modulus of elasticity and W' is the combined load due to the flywheel, rotor and magnetic pull. The allowable deflection depends on the type of generator but should not exceed 0.03 inch under any conditions.

Size of Journals. — The bearing pressure in the main-bearings is determined by the piston pressure, the weight of the flywheel and the belt pulls. In case of a direct-connected machine the weight of the rotor and the magnetic pull must be considered. The pressure on the bearings due to the gas pressure in the cylinder is not in the same direction as the pressure due to the flywheel weight in the horizontal engine. In vertical engines the pressures are all vertical and should be added and it is customary to add them in the horizontal engine also. The main-bearing on the side from which the power is taken off is the one most heavily loaded. The length of the bearing depends upon the rapidity with which the heat of friction may be dissipated. Hence the length depends upon the mean pressure rather than upon the maximum pressure. This is not the mean effective pressure but the mean pressure measured from the atmospheric

line of the indicator card for each stroke. The inertia of the moving parts must also be considered on the suction and exhaust strokes of a single-acting four-cycle engine. During the suction and exhaust strokes the mean pressure may be regarded as the mean height of the inertia curve regardless of whether it is positive or negative, as the suction or exhaust pressure would be added to the inertia for half the stroke and subtracted for the other half. If the mean pressure on the four strokes should be as follows:

Admission,	$p_1 =$	7	pounds	per	square	inch,
Compression,	$p_2 =$	24	"	"	"	"
Expansion,	$p_3 =$	110	"	"	"	"
Exhaust,	$p_4 =$	7	"	"	"	"
Total		148	"	"	"	"

the mean pressure would be 37 pounds per square inch $= p_m$. Then the total mean load due to the cylinder pressure on the main bearing at the right, Fig. 234, would be

$$\frac{P_m}{2} = \frac{p_m \times \text{area piston}}{2}. \tag{1}$$

The total load causing friction would be

$$Q = \frac{P_m}{2} + V_2, \tag{2}$$

where V_2 is the reaction due to all loads between bearings 2 and 3. The work of friction per second per square inch of projected area of the bearing would be

$$F_2 = \frac{Q\pi\, d_2 N \times 0.05}{d_2 \times l_2 \times 12 \times 60} \text{ foot-pounds per second,} \tag{3}$$

where 0.05 is the coefficient of friction and d_2 and l_2 are the diameter and length respectively of the bearing at 2 in inches. This should not exceed 70 foot-pounds per square inch of projected area per second. The same computation should be made for the crank-pin which carries only the total pressure due to the piston pressure. Then

$$F = \frac{P_m \pi\, dN \times 0.05}{d \times l \times 12 \times 60} \text{ foot-pounds per second,} \tag{4}$$

where d is the diameter and l the length of the crank-pin in inches. This value should not exceed 60 foot-pounds per square inch per second.

The pressure per square inch on the crank-pin due to the explosion pressure and on the main-bearing due to the explosion pressure and the reaction of weight s and rope drive should be found. These values should not exceed 1500 pounds per square inch for the crank-pin and 600 pounds per square inch for the main-bearing.

The length of the bearing at 2, Fig. 234, should be found from equation (3) by putting in the value of 70 for F. Then every other term of the equation is known and l_2 may be found. This value of l_2 should then be used to find new values of a, c, etc., in Fig. 234 and a new diameter of shaft computed if the values of a, c, etc., differ greatly from the original assumed values.

It will be noticed that d_2 and d in equations (3) and (4) cancel. This leaves l_2 and l as the only unknowns in these equations which may be used to find the approximate length of bearing and crank-pin at the beginning of the shaft calculations.

Main-bearings. — The main-bearings of an engine are as important as the shaft and should be constructed with the same care given the shaft. There is nothing about the bearings that can be designed rationally after the diameter and length have been computed. The general types of bearings in use are:

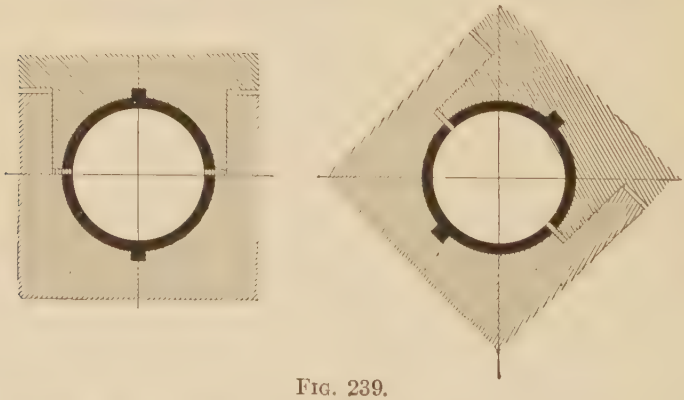


Fig. 239.

Bearings with horizontal or inclined cap such as are illustrated in Fig. 239. The cap is adjustable by means of shims to take up wear. This form of bearing is used only in small engines.

Self-lubricated bearings where lubrication is secured by means of an oil-ring as in Figs. 240 and 241. The latter is a bearing used on the Mietz and Wiess oil engine. This bearing is bolted

to the frame of the engine. The figure also illustrates an efficient system of crank-pin lubrication.

Four-part bearings with wedge adjustment similar to those used in steam engine practice. This type of bearing is illustrated

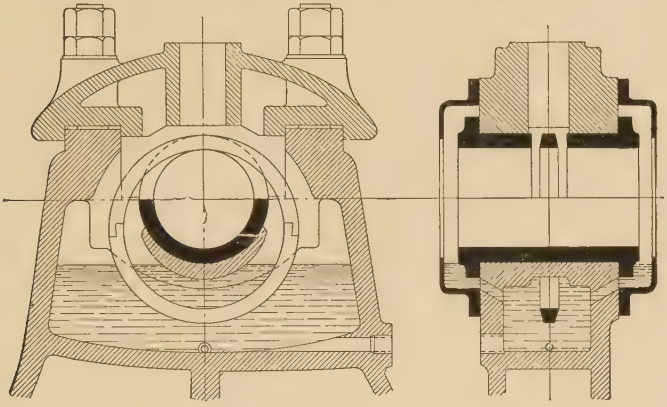


Fig. 240. — Ring-oiled Bearing.

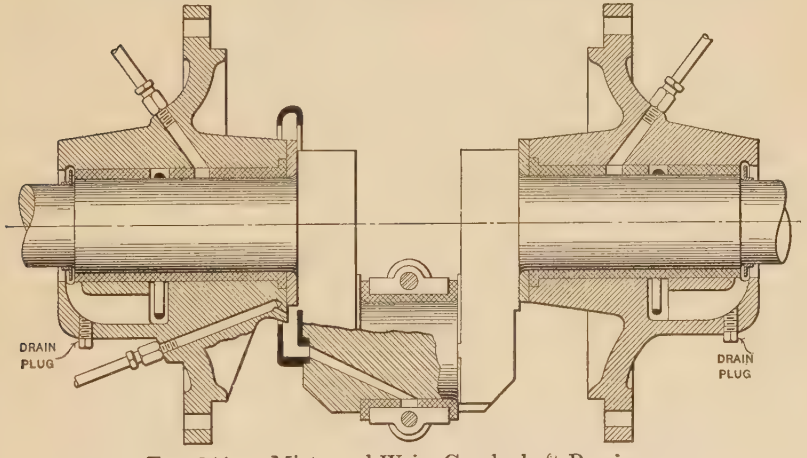


Fig. 241. — Mietz and Weiss Crank-shaft Bearings.

in Fig. 242 which shows a bearing for a large double-acting gas engine. This particular bearing has two wedges for horizontal adjusting. In many cases there is only one wedge fitted for this purpose. Vertical adjustment is taken care of by shims. This bearing also illustrates the water-cooling principle applied to large bearings.

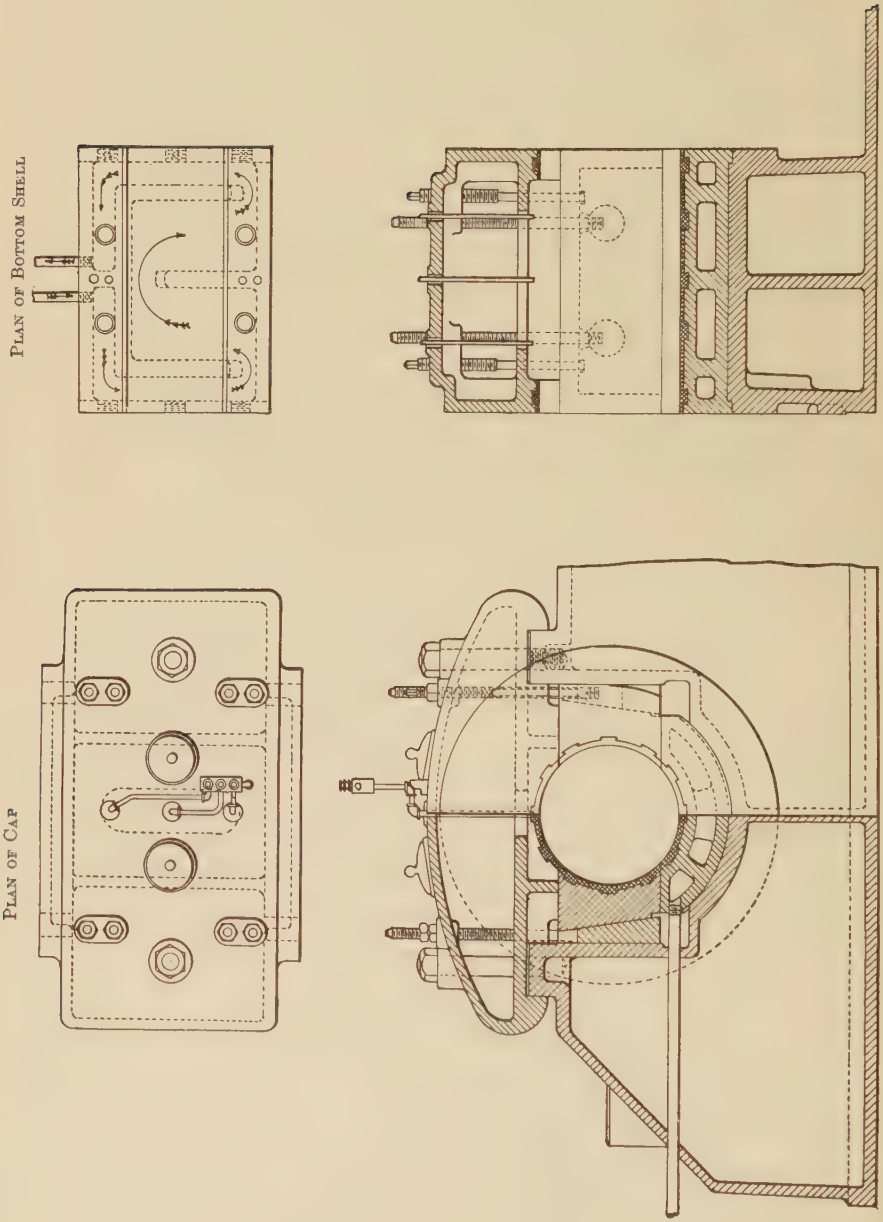


Fig. 242. — Four-part Bearing for Large Gas Engine.

The bed-plate, bearings and shaft for a two-cylinder four-cycle, single-acting Diesel engine are shown in Fig. 243.

The material used for lining the shells of crank-shaft bearings

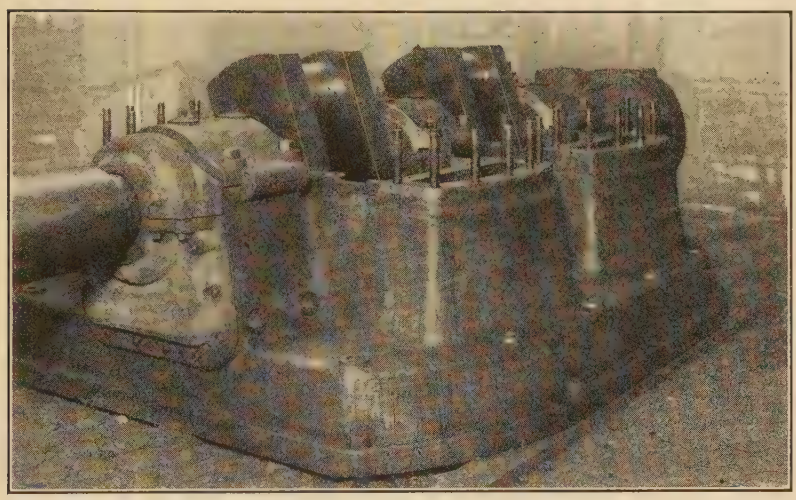


Fig. 243. — Bearings and Crank-shaft for a Vertical Diesel Engine.

is almost always babbitt metal. The shells themselves are made of cast-iron or cast-steel. The thickness of the babbitt depends on the diameter of the shaft, varying from $\frac{3}{16}$ to $\frac{1}{2}$ inch. The babbitt is held in place by means of dove-tail grooves, as shown

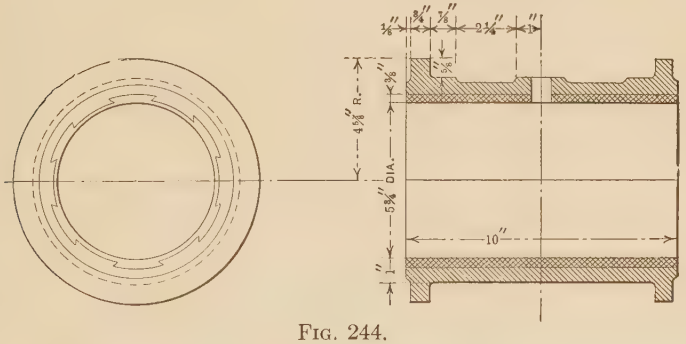


Fig. 244.

in Fig. 244 which illustrates the bearing shells for a $5\frac{3}{4}$-inch shaft for a 10 × 18-inch horizontal oil engine.

Example. — In order to illustrate the methods of shaft calculation used in this chapter there will be made the calculations for

a shaft for a 16 × 24-inch single-acting, four-cycle single-cylinder engine, 200 r.p.m. The mean effective pressure to be 70 pounds per square inch, maximum or explosion pressure 375 pounds per square inch and maximum brake horse-power 72. The type of shaft used will be that shown in Fig. 234. The width of belt required will be about 11 inches with a thickness of $\frac{7}{32}$ inch, making the face of the flywheel 12 inches, the diameter of the wheel being about 9 feet. The tension in the tight side of the belt will be

$$T_1 = \tfrac{7}{32} \times 11 \times 400 = 960 \text{ pounds}, \tag{1}$$

where 400 is the working tension per square inch of belt. The difference in tensions will be

$$T_1 - T_2 = \frac{72 \times 550}{90} = 440 \text{ pounds}, \tag{2}$$

where 550 = foot-pounds per second in one horse-power and 90 is the velocity of the belt in feet per second. From (1) and (2) we get $T_2 = 520$ pounds. The sum of the tensions will then be

$$T_1 + T_2 = 960 + 520 = 1480 \text{ pounds}. \tag{3}$$

The total weight of the flywheel W we will assume to be 12,000 for use in this problem.

Crank-pin. — The distance n may be assumed to be twice the cylinder diameter or 32 inches. The total piston pressure with the crank on the inner center is

$$P = \frac{\pi D^2}{4} \times 375 = 75{,}400 \text{ pounds}. \tag{4}$$

Assuming a and a' equal, the reactions H_1 and H_2 will be equal, 37,700 pounds each. The bending moment at the center of the crank-pin will then be

$$M_b = 37{,}700 \times 16 = 603{,}200 \text{ inch-pounds}.$$

The diameter will be

$$d = \sqrt[3]{\frac{603{,}200 \times 32}{12{,}000 \times \pi}} = 8 \text{ inches}. \tag{5}$$

Allowing 1500 pounds pressure per square inch of projected area, based on the explosion pressure, we find the length of the crank-pin to be

$$l = \frac{75{,}400}{8 \times 1450} = 6\tfrac{1}{2} \text{ inches}. \tag{6}$$

Stress in Left-hand Crank-arm. — The thickness of the crank-arms will be

$$e = 0.65 \times 7.75 + 0.25 = 5\tfrac{1}{4} \text{ inches.} \tag{7}$$

The width will be

$$f = 1.125 \times 7.75 + 0.5 = 9\tfrac{1}{4} \text{ inches.} \tag{8}$$

The distance b is then $\frac{6.5}{2} + \frac{5.25}{2} = 5\frac{7}{8}$ inches and the distance $a - b$ will be $10\frac{1}{8}$ inches. The bending moment on the left-hand arm is then

$$M_b = 37{,}700 \times 10.125 = 382{,}000 \text{ inch-pounds,}$$

and the stress will be

$$K_b = \frac{6 \times 382{,}000}{5.25 \times 5.25 \times 9.25} = 9000 \text{ pounds per square inch.} \tag{9}$$

The stress in the **right-hand crank-arm** will be the same.

Shaft Under Flywheel. — Before the moments under the flywheel may be computed it is necessary to find c, c', the length of bearing $2 = l_2$, and the length of bearing $3 = l_3$. In this type of shaft we may assume the length of all the main-bearings to be the same. In this case $l_2 = \left(a' - \frac{l}{2} - e\right) 2 = 15$ inches. The face of the flywheel is 12 inches. The bending moment due to the weight of the flywheel will be

$$M_w = \frac{12{,}000}{2} \times 21 = 126{,}000 \text{ inch-pounds}$$

and that due to the belt pull will be

$$M_b = \frac{1480}{2} \times 21 = 15{,}120 \text{ inch-pounds.}$$

The combined bending moment will be

$$M_{\text{total}} = \sqrt{\overline{126{,}000}^2 + \overline{15{,}120}^2} = 127{,}000 \text{ inch-pounds.}$$

The diameter to correspond will be

$$d_w = \sqrt[3]{\frac{32 \times 127{,}000}{\pi \times 6000}} = 6 \text{ inches.} \tag{10}$$

Crank at Angle of Maximum Twisting Moment. — It was found by graphical methods that for a length of connecting-rod $5\frac{1}{2}$ times the crank radius the force along the rod at the time maximum twisting occurs is 56,800. Resolving this force parallel

and normal to the crank we find $P_r = 45{,}400$ pounds and $P_t = 38{,}200$ pounds. Referring to Fig. 235 we find

$$H_{t_1} = H_{t_2} = 19{,}100 \text{ pounds},$$
$$H_{r_1} = H_{r_2} = 22{,}700 \text{ pounds},$$
$$V_2 = V_3 = 6000 \text{ pounds}.$$

Shaft Under Flywheel. — The combined bending moment under the flywheel is the same as before, 127,000 inch-pounds. The twisting moment is

$$M_t = 38{,}200 \times 12 = 458{,}000 \text{ inch-pounds}.$$

If this is combined with the bending moment by Guest's law the equivalent twisting moment will be

$$M_{t_e} = \sqrt{458{,}000^2 + 127{,}000^2} = 473{,}000 \text{ inch-pounds}.$$

The diameter to correspond is

$$d_w = \sqrt[3]{\frac{16 \times 473{,}000}{\pi \times 4000}} = 8\tfrac{1}{2} \text{ inches}. \qquad (11)$$

Shaft at Juncture of Right-hand Crank-arm. — The twisting moment here is the same as that under the flywheel. The bending moment is

$$M_b = \frac{56{,}800}{2}\left(16 + 5\tfrac{7}{8} + \frac{5.25}{2}\right) - 56{,}800\left(5\tfrac{7}{8} + \frac{5.25}{2}\right)$$
$$= 215{,}700 \text{ inch-pounds}.$$

The equivalent twisting moment is

$$M_{t_e} = \sqrt{458{,}000^2 + 215{,}700^2} = 507{,}000 \text{ inch-pounds}.$$

The diameter to correspond is

$$d_2 = \sqrt[3]{\frac{16 \times 507{,}000}{\pi \times 6000}} = 7\tfrac{1}{2} \text{ inches}. \qquad (12)$$

The stress used here is higher than that used under the flywheel for two reasons; the condition of loading which was assumed at this point is more severe than the actual condition, and the deflection under the flywheel must be kept small.

Crank-pin. — The bending moment is

$$M_b = \frac{56{,}800}{2} \times 16 = 455{,}000 \text{ inch-pounds},$$

and the twisting moment is

$$M_t = 19{,}100 \times 12 = 229{,}000 \text{ inch-pounds}.$$

The equivalent twisting moment is

$$M_{t_e} = \sqrt{455{,}000^2 + 229{,}000^2} = 511{,}000 \text{ inch-pounds.}$$

The diameter to correspond is

$$d = \sqrt[3]{\frac{511{,}000 \times 16}{\pi \times 5000}} = 8 \text{ inches.} \qquad (11)$$

Left-hand Crank-arm at Juncture of Pin. — The bending moment due to the radial component is

$$M_{b_r} = 22{,}700 \times 10.125 = 230{,}000 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_r} = \frac{6 \times 230{,}000}{5.25 \times 5.25 \times 9.25} = 5420 \text{ pounds per square inch.}$$

The bending moment due to the tangential component is

$$M_{b_t} = 19{,}100\,(12 - 4) = 152{,}800 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_t} = \frac{6 \times 152{,}800}{5.25 \times 9.25 \times 9.25} = 2050 \text{ pounds per square inch.}$$

The stress in direct compression is

$$K_c = \frac{22{,}700}{5.25 \times 9.25} = 430 \text{ pounds per square inch.}$$

The total compressive stress is

$$K_{c_{\text{total}}} = 5420 + 2050 + 430 = 7900 \text{ pounds per square inch.}$$

The twisting moment on the arm is

$$M_t = 19{,}100 \times 10.125 = 193{,}500 \text{ inch-pounds,}$$

and the stress to correspond is

$$K_s = \frac{9 \times 193{,}500}{2 \times 9.25 \times 5.25 \times 5.25} = 3420 \text{ pounds per square inch.}$$

The total combined stress in the arm will be

$$K_{\max} = \frac{7900}{2} + \sqrt{\frac{7900^2}{4} + 3420^2} = 9200 \text{ pounds per square inch.} \qquad (12)$$

Right-hand Crank-arm. — The bending moment due to the radial component is

$$M_{b_r} = 22{,}700 \times 21.875 - 45{,}400 \times 5.875 = 232{,}000 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_r} = \frac{6 \times 232{,}000}{5.25 \times 5.25 \times 9.25} = 5460 \text{ pounds per square inch.}$$

The bending moment due to the tangential component is a maximum where the arm joins the shaft and is

$$M_{b_t} = 38{,}200 (12 - 3.75) + 19{,}100 \times 3.75 = 386{,}600 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_t} = \frac{6 \times 386{,}600}{5.25 \times 9.25 \times 9.25} = 5160 \text{ pounds per square inch.}$$

The direct stress in compression is the same as in the other arm, 430 pounds per square inch. The total compressive stress on one corner is

$$K_{c_{tot}} = 5460 + 5160 + 430 = 11{,}050 \text{ pounds per square inch.}$$

The twisting moment on the arm is

$$M_t = 19{,}100 \times 21.875 - 38{,}200 \times 5.875 = 196{,}000 \text{ inch-pounds.}$$

The twisting stress is

$$K_s = \frac{9 \times 196{,}000}{2 \times 9.25 \times 5.25 \times 5.25} = 3460 \text{ pounds per square inch.}$$

The total combined stress in the arm will be

$$K_{max} = \frac{11{,}050}{2} + \sqrt{\frac{11{,}050^2}{4} + 3460^2} = 12{,}045 \text{ pounds per square inch.} \qquad (13)$$

Length of Bearing at 2. — The bearing at 2 has the largest total reactions, hence this particular one should be designed for bearing pressure. The mean reactions there are due to the mean pressure exerted by the pressure in the cylinder and the weight of reciprocating parts, one-half the flywheel weight and one-half the belt pull. The mean reaction due to the piston pressure and reciprocating parts should be found from the indicator card and the inertia diagram for each stroke and the mean of the four used. In this case we will assume the mean pressure of the four strokes of the cycle to be 40 pounds per square inch of piston. The total reaction at 2 will then be

$$Q_2 = \frac{40 \times 201 + 1480 + 12{,}000}{2} = 10{,}760 \text{ pounds.}$$

The diameter of the shaft at this point, bearing 2, was found to be $7\frac{1}{2}$ inches, equation (12). From the length of crank-pin and thickness of arm we may find the length of bearing 2. Thus

$$\frac{l_2}{2} = a' - \frac{1}{2} - e = 16 - \frac{6.5}{2} - 5.25 = 7\tfrac{1}{2} \text{ inches},$$

from which we find the total length of l_2 to be 15 inches. The work of friction per square inch of projected area per second is

$$F = \frac{10{,}760 \times \pi \times 7.5 \times 200 \times 0.05}{7.5 \times 15 \times 12 \times 60} = 31.3 \text{ foot-pounds}.$$

As this value is far below the maximum value of 70 we will investigate the total pressure on bearing 2. This is

$$R_2 = \frac{201 \times 375 + 1480 + 12{,}000}{2 \times 7.5 \times 15} = 395 \text{ pounds per square inch}.$$

Since this value may run up to 600 pounds per square inch and since the work of friction is low the length l_2 may be reduced. If we make $l_2 = 12$ inches then a', Fig. 235, will be

$$a' = \frac{6.5}{2} + 5.25 + 6 = 14\tfrac{1}{2} \text{ inches}$$

instead of 16 inches as was first assumed. The calculations should be gone over again in exactly the same manner and new diameters and new stresses found as before. In order to avoid tiresome repetition this recalculation is omitted here.

Bending Moment in Bearing 2. — Using the dimensions as in the previous calculations we may find the bending at bearing 2 by the theorem of three moments. Referring to Fig. 236 the values of K' and K'' will each be 0.5. The load P' between 1 and 2 will be 56,800 which we may assume to be in a horizontal direction. The load W between bearings 2 and 3 is 12,000 pounds, neglecting the belt pulls. The bending moment at bearing 2 due to P' is

$$M_{2p} = -\frac{56{,}800\,(0.5 - 0.5^3)\,\overline{32}^2}{2\,(32 + 42)} = -147{,}500 \text{ inch-pounds},$$

$$R_{p_1} = -\frac{147{,}500}{32} + 56{,}800\,(1 - 0.5) = 23{,}780 \text{ pounds},$$

$$R_{p_3} = -\frac{147{,}500}{42} = -3510 \text{ pounds},$$

$$R_{p_2} = 56{,}800 - 23{,}780 + 3510 = 36{,}530 \text{ pounds}.$$

The bending moment at bearing 2 due to W is

$$M_{2w} = -\frac{12{,}000 \times \overline{42}^2\,(2 \times 0.5 - 3 \times 0.25 + 0.125)}{2\,(32 + 42)}$$

$$= -53{,}700 \text{ inch-pounds},$$

$$R_{w_1} = -\frac{53{,}700}{32} = -1675 \text{ pounds},$$

$$R_{w_3} = 12{,}000 \times 0.5 - \left(-\frac{53{,}700}{42}\right) = 7280 \text{ pounds},$$

$$R_{w_2} = 12{,}000 - 7280 + 1675 = 6395 \text{ pounds}.$$

The twisting moment on the shaft is

$$M_t = 38{,}200 \times 12 = 458{,}000 \text{ inch-pounds}.$$

The bending moment at the center of the crank-pin due to P' is

$$M_p = R_{p_1} \times a = 23{,}780 \times 16 = 380{,}000 \text{ inch-pounds}.$$

The bending moment under W due to W is

$$M_w = R_{w_3} \times c_1 = 7280 \times 21 = 153{,}000 \text{ inch-pounds}.$$

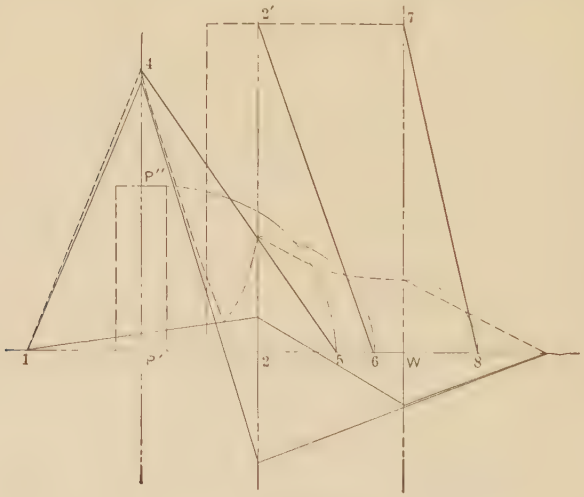


FIG. 245.

The bending moment, twisting and combined diagrams are shown in Fig. 245. The moments due to the piston pressure and flywheel being at right angles to each other are combined by finding the square root of the sum of their squares. The combined bending moments are shown by the dotted broken line. At the

center of the crank-pin P', at bearing 2 and at the center of the flywheel W the combined bending moments are combined with the twisting moments at those points, resulting in equivalent twisting moments 4–5 at the crank-pin, 2′–6 at the center of bearing 2 and 7–8 at the center of the flywheel. These moments have the following values:

$$4\text{–}5 = 469{,}000 \text{ inch-pounds},$$
$$2'\text{–}6 = 483{,}000 \text{ inch-pounds},$$
$$7\text{–}8 = 466{,}000 \text{ inch-pounds}.$$

In the computations for this shaft as a simple beam it was found that the combined bending and twisting moment at the center of the crank-pin was 511,000 inch-pounds, that at the place where bearing 2 joins the crank-arm was 507,000 inch-pounds and that under the flywheel was 473,000 inch-pounds. Thus the shaft is safe as already designed.

Five-bearing Crank-shaft. — The five-bearing crank-shaft for a four-cylinder vertical engine with an overhung flywheel is shown in Fig. 246. This shaft should be investigated for at least 3 conditions of loading as follows:

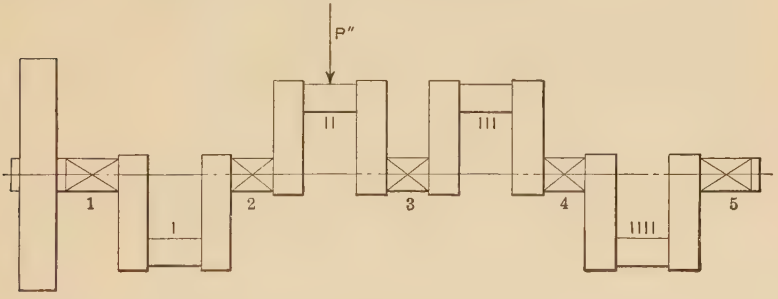


Fig. 246.

First, with the second crank from the left on the top center at the beginning of the explosion stroke, assuming the engine to be just starting up. Then the load on this crank is the explosion pressure multiplied by the area of the piston and the loads on the other cranks are zero.

Second, with the same crank at the angle of maximum twisting moment. In this position the load on this crank should be found as in the previous problem. The loads on the other cranks will be practically zero.

Third, with the cranks in position given under the first condition but with the engine up to speed. The load P'', Fig. 247, will be the explosion pressure minus the force of inertia, multiplied by the area of the piston. The other loads will be the inertia in each case multiplied by the area of the piston. P''' will be upward, P' and P'''' will be downward.

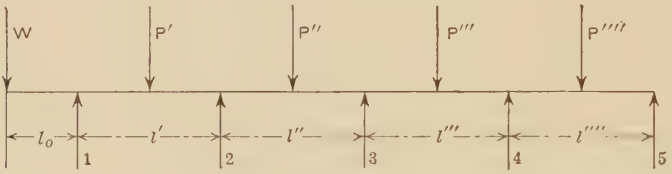


Fig. 247.

In all of the above cases the bending moments should be found by the theorem of three moments and plotted as was done for the three-bearing shaft. In the second case there will be twisting in addition to the bending.

The moments may be found by the following equations. The spans are all the same length except l_0 and the loads are in the middle of the spans in every case except the flywheel which is an overhung load. The subscripts of the bending moments refer to the bearing numbers.

$$M_1 = -Wl_0,$$

$$M_1l' + 2(l' + l'')M_2 + M_3l'' = -P'l'^2(\tfrac{1}{2} - \tfrac{1}{8}) - P''l''^2(1 - \tfrac{3}{4} + \tfrac{1}{8}),$$

$$M_2l'' + 2(l'' + l''')M_3 + M_4l''' = -P''l''^2(\tfrac{1}{2} - \tfrac{1}{8}) - P'''l'''^2(1 - \tfrac{3}{4} + \tfrac{1}{8}),$$

$$M_3l''' + 2(l''' + l'''')M_4 + M_5l'''' = -P'''l'''^2(\tfrac{1}{2} - \tfrac{1}{8}) - P''''l''''^2(1 - \tfrac{3}{4} + \tfrac{1}{8}),$$

$$M_5 = 0.$$

The shear at the right of the various supports will be as follows:

$$V_1 = \frac{M_2 - M_1}{l'} + \frac{P'}{2},$$

$$V_2 = \frac{M_3 - M_2}{l''} + \frac{P''}{2},$$

$$V_3 = \frac{M_4 - M_3}{l'''} + \frac{P'''}{2},$$

$$V_4 = \frac{M_5 - M_4}{l''''} + \frac{P''''}{2}.$$

The moments under the loads will then be

$$M' = M_1 + \frac{V_1 l'}{2},$$

$$M'' = M_2 + \frac{V_2 l''}{2},$$

$$M''' = M_3 + \frac{V_3 l'''}{2},$$

$$M'''' = M_4 + \frac{V_4 l''''}{2}.$$

EXERCISES

1. Design the crank-arm for a 14×24 gas engine, explosion pressure 375 lbs. per sq. in. Referring to Fig. 229 assume $l_1 = 7$ ins., diameter of shaft at bearing $1 = 8$ ins. Maximum twisting moment on shaft occurs at a crank angle of 35 degrees at which time there is a pressure on the piston of 125 lbs. per sq. in. The connecting-rod of the engine is 65 ins. long.

2. Find the bending moments at bearing 1 and under the flywheel of the above engine when the crank is on the dead center. Assume $c = c' = 23$ ins., $W = 20{,}000$ lbs., $l = 16$ ins., $e = 6$ ins. and $T_1 + T_2 = 3000$ lbs.

3. Find the bending and twisting moments at 1 and 2 when the crank is in the position of maximum twisting moment as given in Prob. 1.

4. Referring to Fig. 234 assume $P = 75{,}000$ lbs., $r = 10$ ins., $a = a' = 15$ ins., $b = 9$ ins., $e = 5\frac{1}{2}$ ins., $f = 9$ ins., $W = 16{,}000$ lbs., $c = 22$ ins., $c' = 18$ ins. and $T_1 + T_2 = 3000$ lbs. Find H_1, H_2, V_2, and V_3. Find the bending moments at the center of the crank-pin and at W.

5. Using the data given in Prob. 4 find the stress in the left-hand crank-arm when the crank is at an angle of 35 degrees, pressure on the piston 150 lbs. per sq. in. Length of connecting-rod 60 ins. Dia. cyl. = 16 ins.

6. Using the conditions given in Probs. 4 and 5 find the diameter of the crank-pin.

7. Using the conditions given in Probs. 4 and 5 find the diameter of the shaft under the flywheel.

8. Referring to Fig. 247, assume $l' = l'' = l''' = l'''' = 15$ ins., $l_0 = 11$ ins., $W = 3000$ lbs., $P' = 1000$, $P'' = 18{,}000$, $P''' = 1200$ and $P'''' = 2000$ lbs. Find the bending moments at bearings 1, 2, 3, 4 and 5 and at each load. Draw the bending moment diagram.

CHAPTER XXIII

THE HUMPHREY GAS PUMP

The *Humphrey* gas pump was invented by Herbert A. Humphrey of England. Its outward appearance has no resemblance to an internal combustion engine; it is only in operation that the two are similar. Its adoption in England and Continental Europe has been very slow and only a few minor installations have been made in the United States. Hence it cannot be called successful, commercially, and is included in this work only because it is of interest as a part of the general internal combustion problem.

For heads up to 50 feet the gas pump is very simple — a piece of pipe with a stand-pipe at one end and a combustion chamber at the other, the latter being surmounted by a few simple fittings. Fig. 248 is a diagram of the simple pump. It consists of a conical

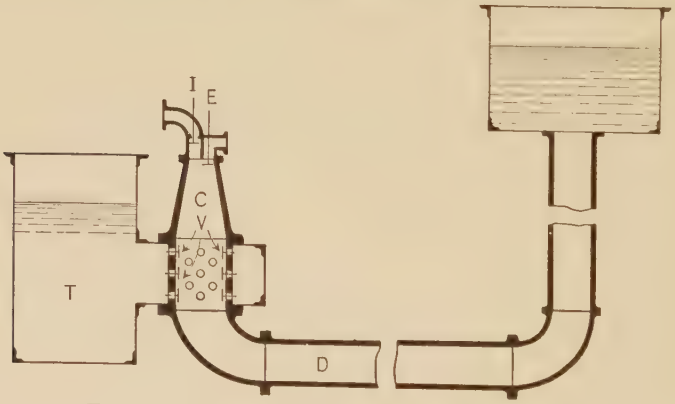


FIG. 248. — Diagram of Simple Humphrey Pump.

explosion chamber *C*, fitted at the top with inlet valve *I*, the exhaust valve *E*, and a scavenging-valve not shown. A simple interlocking gear is arranged between these valves by means of which, when valve *I* has opened and closed, it is locked in the closed position and valve *E* is released. When valve *E* has opened and closed it is locked and valve *I* is released.

Imagine a charge of gas and air compressed in the top of chamber *C* and fired by a spark plug which projects through the top of the casing. All the valves are shut when explosion occurs, and the increase in pressure drives the water downward in the

pump and sets the whole column of water in the discharge pipe in motion. The column of water attains kinetic energy while work is being done on it by the expanding gases, so that when these gases reach atmospheric pressure the column of water may be moving at, say, 6 feet per second. The motion of this column of water continues until the pressure behind it falls far enough below the atmosphere to open the exhaust-valve *E* and the water-valves *V*. Water rushes in through these water-valves and follows the moving column in pipe *D*, and at the same time it rises in chamber *C* in effort to reach the level in the suction tank.

When the kinetic energy of the moving column has been expended in forcing water into the high level tank, it comes to rest and, there being nothing to prevent a return flow, the column starts to move backward toward the combustion chamber and continues with increasing velocity until the water reaches the level of the exhaust valve, which it shuts by impact. A certain part of the burned gas is trapped in the cushion space beneath the inlet valve, and the energy of the moving body of water is expended in compressing this burned gas to a greater pressure than that due to the static head of the water in the tank *T*. A second outward movement of the column now begins and when the water reaches the level of the valve *E*, the pressure in the compression space is again atmospheric and further movement of the water opens the valve *I*, which has been released by the interlocking device, thus drawing in a fresh charge of gas and air. Once more the column of water returns under the pressure of the elevated tank and compresses the charge of gas and air, which is then ignited to start a new cycle of operations.

Ignition. — Ignition is timed by an apparatus resembling an ordinary engine indicator, which closes the electric ignition circuit at the point of maximum compression, and an ordinary small battery, trembler coil and spark plug, such as are used in automobile work, are employed.

Starting. — In starting the pump for the first time compressed air is allowed to flow into the combustion chamber until the volume of the air thus introduced is slightly larger than the usual charge. The exhaust valve is then suddenly opened by means of a hand lever, and the escape of the compressed air permits a movement of the water column which gives the cushion and suction strokes, and so draws in a fresh combustible charge, which, on ignition,

starts the pump working regularly. If the pump is stopped when working regularly, it always stops with a fresh charge in the combustion chamber, under pressure, so that the only operation necessary for starting is to close the switch on the ignition circuit. The pump can therefore be started and stopped from a switchboard.

Two-cycle Pump. — The Humphrey pump described in the preceding articles works on the four-cycle principle, requiring four strokes for completion of one cycle. The time required for the completion of a cycle is much greater than in an ordinary engine and is not governed in design except by the length of the "play pipe" *D* in Fig. 248. Fig. 249 shows an indicator card taken from one pump in which the time per cycle was about 5 seconds, or actually the rate was 12.5 cycles per minute.

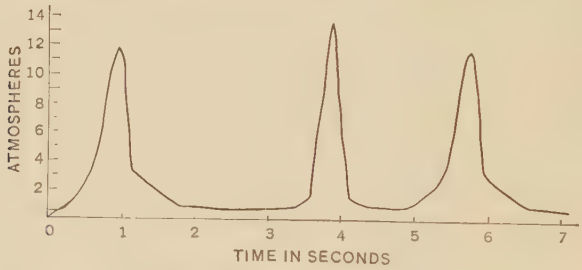


FIG. 249. — Indicator Diagram from Humphrey Pump.

In Fig. 250 is illustrated a double-cylinder pump which draws water twice each cycle, and, by means of the air chamber called the "intensifier," placed between the combustion chamber and a set of delivery valves, will deliver water against a head which is independent of the mean pressure of explosion. *A* and *B* are the two combustion chambers, *C* and *D* two water valve boxes, *E* the connecting chamber and *F* the discharge pipe. In the top of *A* and *B* are fitted the usual inlet and exhaust valves, but in this case the exhaust valve is set near the top of the chamber as practically no cushion is wanted. The pump works with a drowned suction, being placed inside the tank *T*, the water supply being through *M*. Starting with a compressed combustible charge in *A* and with *B* full of water, explosion occurs and drives the water in *A* downward and outward, setting the whole column of water in *F* in motion. When the gases in *A* have expanded nearly to atmospheric

pressure, the water column in B is no longer supported and, falling by its own weight, draws in a fresh combustible charge which is throttled to prevent the charge being too large. The inlet water coming in through the valves follows the outward moving column and also rises in chamber A, to expel the burnt-products up to the level of the water in tank T. The column in the discharge pipe, having come to rest, now begins to return, and, rising in chamber A, expels the burnt-products and closes the exhaust valve. The energy of the moving column is now expended in compressing the new combustible charge in B, whereupon ignition in B starts a new cycle with functions of A and B reversed.

Coming to the intensifier, we have the small air chamber G and the larger air vessel K with a valve box J. The small air vessel G is fitted with an adjustable pipe H, open to the atmosphere and

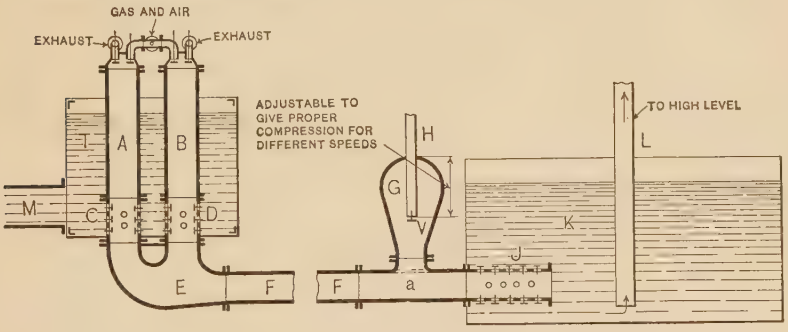


FIG. 250. — Humphrey Two-cycle Pump.

closed at the bottom by a valve which opens under its own weight and closes under the impact of the water on it. When the explosion occurred in chamber A the water level in the small air vessel was, say, a, and while rising from a to V the energy of the expansion in A is wholly used in giving kinetic energy to the water column. As the water strikes V it shuts this valve and then compresses air in G until the pressure equals that in the high-pressure air vessel K. The valves in J now open and the energy in the water column is expended in driving water through these valves into K. When the water column stops moving, the valves in J close and the energy of the compressed air in G drives back the water column, exhausts the burnt-products from A, and compresses the fresh combustible in B. Thus everything is ready for the next cycle.

It will be seen that by raising the pipe H the quantity of air compressed in G is lessened, and by lowering H the quantity is increased. Thus one has complete control over the amount of energy which is stored in G and subsequently utilized for compression of the fresh charge. This enables one to operate with any compression pressure that may be desired. It also enables one to keep the compression pressure constant in A and B, no matter what may be the pressure of delivery of the water from K through the delivery pipe L. Thus, if the head is raised, pipe H can be raised, and this is the only adjustment necessary to make in pumping against different heads.

At the end of the expansion stroke the gases in the combustion chamber of the Humphrey pump are so cool that fresh mixture may be drawn in and come in contact with the exhaust gases without fear of premature explosion. Fig. 251 shows the combustion chamber of a two-cycle Humphrey pump designed in accordance with the above fact. The combustion chamber has to be specially shaped so that the incoming charge, which may be preceded by pure air, displaces the burnt-products and mixes as little as possible with them. In the sketch A is the admission valve at the top of the tall but narrow part of the chamber B in which the full charge volume extends down to the line cc. A number of exhaust valves E lead to a common outlet O which may be fitted with a non-return valve, or each exhaust valve may carry a light non-return valve on its spindle as shown. The level at which the expansion reaches atmospheric pressure is, say, ff, but this level having been reached by the water its further movement draws in a fresh combustible mixture until this mixture occupies the space down to cc, and the liquid level has fallen to gg. The column of liquid now returns and drives the exhaust products through the valves E (which open by their own weight) until these valves are shut by the water. The kinetic energy acquired by the column is now spent in compressing the fresh charge, which is ignited to start a new cycle. Thus, each out-stroke is a working stroke and no locking gear is required on the valves.

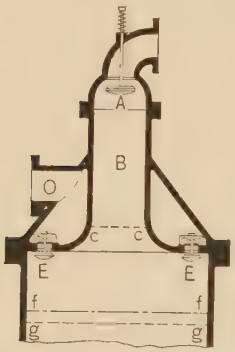


FIG. 251.—Combustion Chamber. Humphrey Two-cycle Pump.

The Humphrey Air Compressor. — If the column of water oscillating in the play pipe of a Humphrey pump is used as a water piston and caused to rise and fall in an air vessel fitted with suitable valves for the inlet and outlet of air, the combination constitutes an air compressor of a very efficient type.

A compressor with a single-barrel pump and a single air cylinder is illustrated in Fig. 252. *A* is the ordinary pump chamber and *C* the air compressor chamber. The inlet valve *i* for the gas and air and the exhaust valve *e* exist as before, but the exhaust pipe and valve are adjustable in a vertical direction so as to vary the cushion space at the top of *A*. There is another pipe *p* fitted with a valve *r* and also adjustable vertically. A reservoir *X*, such as a boiler shell, is connected by a flexible pipe to *p*. Corresponding parts are fitted to the compressor; thus, *f* is an

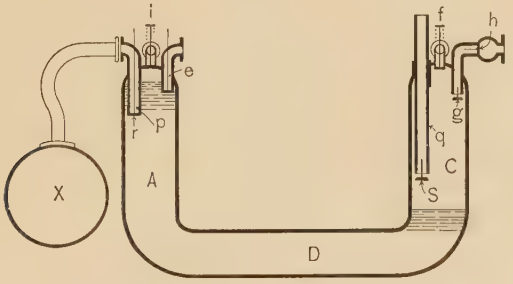


FIG. 252. — Humphrey Simple Air Compressor.

air inlet-valve; *g* an outlet-valve for the air, which is shut by the water; and *h* a non-return valve for the compressed air. The dip-pipe *q* with its valve *S* gives communication between the atmosphere and the compressor. The outlet-pipe and the dip-pipe are adjustable in a vertical direction.

The cycle can be explained by assuming the water level to be high in *A* and low in *C* as shown in the figure, and that a compressed charge is ignited in *A*. Expansion occurs, driving down the water in *A* and up in *C*. At first air escapes from *C* through the valve *S*, but when the water reaches this valve it is shut and the remaining air is trapped and compressed until the pressure of the discharge is reached, when *h* opens and compressed air is delivered.

Next, the water reaches valve *g* and shuts it so that no more air may escape, and the water column is brought to rest by the

continued compression of the remaining air which forms a cushion. Meanwhile in chamber A expansion to atmosphere allows the exhaust valve e to open, and the continued stroke draws in scavenging air (the scavenging valve is not shown), but the valves i and r remain locked. Now the column begins to return, due to the expansion of the cushion air in C, and this causes exhaust and cushion in A and at the same time there is first a fall of pressure in C to atmosphere, then the valve f is opened against a light spring, allowing a fresh supply of air to enter.

So far we have had one outward and one return stroke, but the expansion of the cushion in A starts a second outstroke which results in drawing in a new combustible charge in A through valves i and r, which were released on the closing of the exhaust valve e. As the second outstroke is nearly as long as the first, it follows that too much combustible mixture is drawn in, but the surplus is gotten rid of on the second return stroke by allowing it to escape into the receiver X, thus slightly raising the pressure therein. Valve i is shut by its spring but r is shut by the action of the water, and when this occurs, chamber A contains a definite volume of combustible mixture, the size of the charge being fixed by the height of r. The second outstroke towards C merely compresses the air in C, without causing delivery to take place, but stores in the air the energy given out by the expansion of the cushion in A. The expansion of the stored air in C gives the compression stroke in A and the combustible charge is ready to be ignited to start a new cycle.

By making adjustments on the levels of the exhaust valve e and the rejected-charge valve r, which are so constructed as to make this possible, and similar adjustments on the compressor side, the machine may be suited to variable pressure and air quantity as desired.

It is also practicable to cause the variation in pressure in X to automatically adjust the quantity of water which reciprocates between the chambers to suit the capacity per working cycle, as theory would indicate.

Newer Developments. — One of the newer developments in the Humphrey pump is a device for rendering the compression independent of the head. This device is illustrated in Fig. 253. The chamber at the left represents the combustion chamber of the pump. In this case the water inlet valves are provided near

the delivery end of the pipe as shown at A. The chamber B is an air vessel from which the water may be delivered against a high head. A valve C closes the end of the delivery pipe. On the outstroke this valve is raised by motion of the water, some of which thus escapes into the air chamber. On the return stroke this valve, following the motion of the water, closes the delivery end of the play pipe, and is finally brought quietly to rest by the dash-pot shown at D. The water in the play pipe having, however, already acquired momentum, still moves backward and a fresh supply of water then enters at the inlet-valve. By suitably adjusting the motion of the cut-off valve C any desired proportion of the energy of the cut-off stroke can be utilized in compressing the new charge.

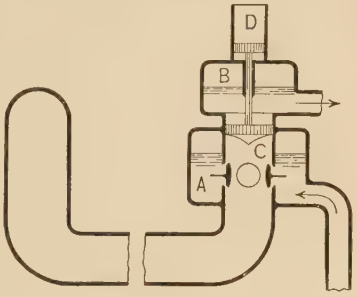


FIG. 253.

A Humphrey pump may also be arranged to have a combustion chamber at each end of the play pipe, as illustrated in Fig. 254,

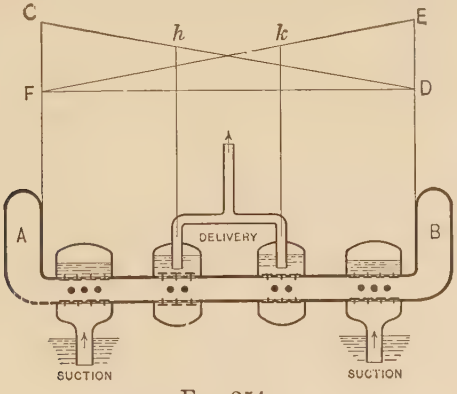


FIG. 254.

where the delivery valves are placed in the region of the pipe where the pressure is most constant. When an explosion occurs in chamber A, and the column of liquid is accelerated, the distribution of pressure along the play pipe is represented by the sloping line CD. At the first delivery box the pressure is represented by h, and so long as this is higher than the back pressure, water

will pass the delivery valves. As the point of delivery approaches nearer and nearer to the combustion chamber, so may the pressure of delivery be increased, and on these lines a pump has been built to pump against a head of 300 feet, including the suction.

A device for increasing the number of cycles per minute has been introduced by Mr. Humphrey and is illustrated in Fig. 255. Referring to Fig. 253, it will be seen that, with the arrangement there illustrated, the water in the play pipe is never changed. Its sole use is by its momentum to provide for the reciprocations required in the combustion chamber. It can therefore be replaced by a mass of metal as indicated in Fig. 255, a device which in certain cases has considerable advantage. In the first place for the same inertia the bulk is reduced to one-seventh of that needed with water, and, secondly, higher rates of retardation and acceleration become possible. With the ordinary type of pump

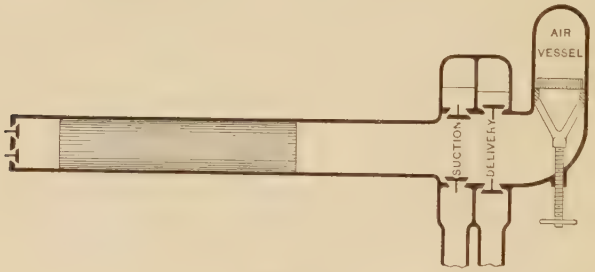


FIG. 255.

it is found that if the retardation on the return stroke is greater than 32 feet per second, water is liable to be sprayed from the surface of the water into the combustion chamber, reducing the thermal efficiency. By using a solid piston this limit to the rate of retardation is eliminated and the rate of reciprocation can be greatly increased. Using a solid piston there is no necessity for making the play pipe horizontal; it may be made vertical without any inconvenience. With an experimental pump of this kind the rate of reciprocation has been increased to 193 cycles per minute.

Efficiency of the Humphrey Pump. — Higher efficiencies than those obtained in the Otto cycle are claimed for the Humphrey pump. Tests made on pumps of this type seem to indicate that the efficiency is actually higher than the efficiency of the reciprocating engine of the same power operating on the same com-

pression. One of the original pumps, a 35 horse-power unit, was tested, and showed efficiencies of 21.25 and 22.07 per cent based on water horse-power. Larger pumps have shown corresponding increase in efficiency.

Advantages of the Humphrey Pump. — The advantages claimed for this pump are as follows:

1. No moving parts required except a few mushroom valves.
2. Practically no lubrication.
3. Simple construction.
4. No cooling jacket.
5. High thermal efficiency.
6. No valves on the delivery side of the pump.
7. No flywheel need be used.
8. Light weight.
9. Low first cost.

CHAPTER XXIV

GAS PRODUCERS

The fact that coal can be converted into gas by the combined influence of air, water vapor, and heat has been known for many years. Gas formed by such methods, and the innumerable modifications that have been proposed, has generally been known as producer gas and has occupied an important place in various industries ever since Siemens applied it to his regenerative furnace in 1861. Economies were found possible by the application of gaseous fuels to furnaces even in the days of cheap fuel. Nearly every industry employing high temperatures, wherever cheap natural gas was not obtainable, used producer gas. It is safe to say that a great majority of all of the steel and glass manufactured throughout the world during the past fifty years was at some stage of its making heated by producer gas.

The apparatus used in generating this gas was simple and crude, but it was sufficient for the purpose and very little improvement was attempted until quite recently. This apparatus consisted for the most part of a steel shell brick lined with water seal bottom for the removal of ash, and a gas offtake through which the gas was driven by means of steam pressure, the induced air entering the generator at the bottom. Coal charged through a bell hopper permitting of a seal to the producer when coal was admitted to the hopper. In this way the oxygen and the hydrogen in the air and steam were combined with the carbon of the coal to form carbon monoxide and hydrogen. Due to the presence of nitrogen the heating value of gas thus made was usually not more than 120 to 130 B.t.u. per cubic foot. The gas itself was used in the hot and crude state just as it passed from the gas generator.

The simple equipment required for the generation of producer gas and the very high thermal efficiency of the process attracted the attention of engineers, and the possibility of extending the use of producer gas to other fields where a hot gas could not be used, brought about methods of scrubbing and cleaning to make

this cheap gas available where otherwise a much more costly fuel would be required.

The internal combustion engine was just at this time coming into prominence and it was very natural that the gas producer was thought of as a source of cheap gas for the generation of power. During the last years of the nineteenth century we learned in the United States of what appeared to be wonderfully simple and exceptionally efficient power plants that were being operated in England and Germany, consisting of a suction gas-producer furnishing gas to a gas engine. The early descriptions current at this time indicated that this equipment consisted of a simple sheet-iron furnace lined with fire brick in which the fuel was placed, with a pipe leading from this furnace to one or two tanks in which a spray of water was used to cool and clean the gas. These steel tanks usually contained broken stone or coke to act as a method of more finely dividing the gas and bringing it in contact with the water. The suction of the engine drew the gas from the producer. It was said that practically no attention was required and a fuel economy of 1 pound per brake-horse-power hour was easily obtained. The chemistry of gas making disclosed at that time was equally elementary and was worked out as follows: $2C + O_2 = 2CO$, $C + 2H_2O = CO_2$ and $2H_2$. Thousands of small power plants were built on this plan and in successful operation throughout Europe, so that this new development received an enthusiastic welcome from American manufacturers of gas-engines, who obtained by various methods the design and manufacturing rights of successful foreign makes of producers to be sold in connection with their several designs of gas engines. The failure of many installations in this country was without doubt due to the lack of knowledge of the apparatus, and also to the fact that many manufacturers in this country failed to provide an adequate technical staff to handle this work.

The fundamental error in these early attempts was the importation of equipment designed for European conditions without any clear understanding of how vitally different were the conditions to be met here. Almost without exception plants of European design failed, and no small factor in this was the radical difference between the fuels obtainable here and those used abroad.

Producer design which originated in this country fared somewhat better since the problem of fuel was encountered during

the experimental or development stages and measures were taken to conform the producer to the fuel. This, in many cases, involved a very radical departure from European practice and compelled the American designer to start on entirely new lines. As a consequence he made mistakes, but since that time has developed a very excellent machine.

Some early failures in this country may also be attributed to a misleading idea which prevailed that inasmuch as licensed engineers were not required they did not necessarily have to be intelligent, and the lack of unskilled operators and men who were familiar with the chemistry of the process often resulted in discontinuing the plant which might otherwise have been a successful and economical installation.

The producer of today is a growth embodying the results of years of experience in a field of a highly organized technical staff of the manufacturer.

The producers used for power work in the early days used only anthracite coal or some similar fuel containing no tar-forming constituents. Many attempts were made to use the more plentiful bituminous coals, lignite, etc., but without any great success. The early efforts to handle bituminous coals were of two distinct lines: First, those plants that undertook to remove the tar from the gas after it left the producer. Second, the plants that undertook to destroy the tar within the producer itself. It is probably safe to say that today any fuel that will burn can be used as a fuel for producer gas for gas engines. The selection of the most suitable fuel for any given installation, however, calls for the careful consideration of all factors involved including the cost of the fuel, its availability, the cost of the particular equipment required to correctly gasify the fuel and clean the gas, labor, building, water, power, etc. Any attempt to formulate a general rule for such cases is impossible, but it may be of general interest to know that gas producers for clean gas are in operation today that range in sizes from a capacity of 3 pounds of fuel per hour to 6000 pounds per hour, but we know of no producers used on power work only that are gasifying over 2000 pounds per hour.

A gas producer will operate on a low grade fuel that cannot be utilized economically in any other way, and it is without doubt one of the best ways to make available the heat of lignite and similar fuels which carry much water. It is easy, however, to

make a mistake in a decision to utilize waste fuels, and in each case it is necessary to determine the economical advisability of such procedure. The same may be said regarding the recovery of by-products. It is possible to recover much ammonia from producer gas, but in most instances the net profit available from such a procedure is more often negative than positive. Inferior fuels always mean larger investment in the gas producer equipment, decreased capacity per unit, increased labor and possibly an extra charge for the removal of ash or other waste from the plant. These charges may be more than the original cost of a good fuel.

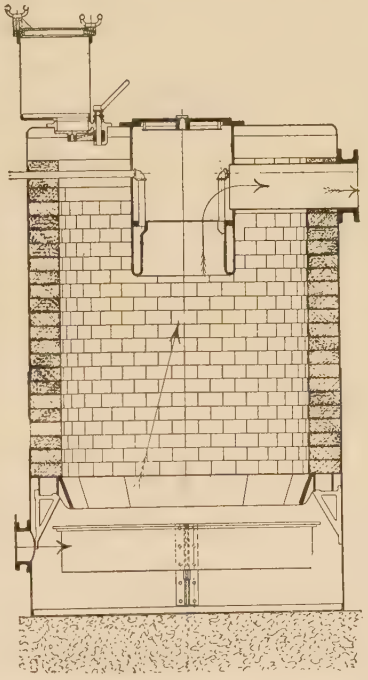


Fig. 256.—Section of Simple Anthracite Producer.

Principles Involved in Gas Producers.—The ideal producer gas for gas-engine work must contain as large an amount of CO as it is possible to obtain, a certain portion of hydrogen, and as little CO_2 as it is possible to make. Inasmuch as the gas-engine acts as a pump for withdrawing the gas from the producer only sufficient air for the conditions encountered is drawn into the bottom of the producer. With the ingoing air is added either steam or hot water so that the air may be saturated to a given temperature and carried into the producer, and a definite amount of water vapor, part of which eventually appears in the form hydrogen in the gas. A close study of the reactions which take place in a gas producer show clearly the absolute necessity of certain features in the manufacture of a gas-producer plant. The reactions in a producer using bituminous coal are more complex than one operating on anthracite coal, so that the following discussion will for the greater part be limited to the reactions in the anthracite machine. Fig. 256 shows a section of a simple

anthracite gas producer and is typical of the round type machines used for gas engine work. Essentially it consists of a cylindrical steel shell closed at the top and brick lined. A grate at the bottom supports the fuel and provision is made for charging fresh fuel through a gas tight hopper at the top. Gas is collected in the center outlet and steam and air are introduced beneath the grate. When in operation the fuel bed is several feet deep and covers the mouth of the gas offtake.

Assuming first that air only, which is composed of oxygen and nitrogen, is admitted beneath the grate, the oxygen of the air unites with the carbon of the coal in the lower part of the fuel bed forming carbon dioxide with the liberation of heat exactly as when coal is burned in a furnace. The hot gases consisting of carbon dioxide and nitrogen pass up through the fuel bed towards the gas outlet giving up most of their heat to the surrounding coal. When a sufficient depth of this coal is heated to a bright red color another reaction takes place between the carbon dioxide and the hot carbon resulting in a reduction of the carbon dioxide to carbon monoxide, the latter being a combustible gas but the nitrogen remains unchanged. If the introduction of air is continued sufficiently long the temperature within the apparatus will rise, the ash will fuse and the linings and grate melt down. To prevent this condition water vapor or steam is introduced with the air. Under the influence of the heat and in the presence of carbon the steam coming in contact with the hot fuel is decomposed into its chemical elements, hydrogen and oxygen. The oxygen unites with the red-hot carbon of the fuel and assists in carrying on combustion. The hydrogen passes through the fire and becomes a desirable part of the producer gas.

The formation of carbon dioxide produces heat and is called a positive reaction, whereas the formation of carbon monoxide by the reduction of carbon dioxide and the breaking up of water into its elements require heat and are called negative reactions.

Too much air not only causes the formation of clinker and rapid deterioration of the apparatus because of excessive temperatures but the fire becomes channeled and the air passes through without coming sufficiently in contact with the fuel to allow the reactions to take place. Too much steam results first in producing a gas very rich in hydrogen and quickly deadens the fire to a point where no reactions at all can take place. The proper

mixture of the two is what maintains the fire in good gas-making condition. It is necessary that the gas should not be of a varying quality. If the gas as supplied to the engine varies, this rapid fluctuation will surely cause trouble. It will be recalled that no steam engine can operate properly with a varying pressure at the throttle, nor can a gas engine operate properly with a gas that is varying rapidly in quality as delivered to the engine.

The results are more difficult to obtain than might appear at first sight and requires several essential features of construction, the most important of these are: first, an automatic, quick-acting device for maintaining the proper proportions of steam and air under all conditions; second, uniform distribution of the air and steam and intimate contact with the incandescent fuel; third, efficient cooling and cleaning apparatus; fourth, capability of the producer to respond to sudden variations in load; fifth, proper design for the fuel to be used with suitable provision for charge of new fuel, and the removal of ash without interfering with the continuous operation of the producer.

Fuel.—The fuel best adapted to the anthracite producer is either No. 1 Buckwheat anthracite or pea coal. This coal should and will usually contain not more than 1 per cent sulphur and not more than 15 per cent ash, while the fuel for bituminous producers for best results should be a non-coking high volatile size coal. This coal should normally run from $\frac{3}{4}$- to 3-in. lump, and although run-of-mine coal is sometimes used it invariably requires more producer capacity to handle a given amount of fuel.

The rate of combustion on the anthracite producer for best results should not exceed 8 pounds per square foot of grate surface. For bituminous producers it may run from 12 to 25 pounds per square foot of grate surface depending upon the type of producer used.

Producer Plant Cost.—It is difficult to make any direct statement as to the cost of a gas-producer gas engine installation, or to make comparisons with prime movers of other types. In general, it may be stated that up to a given size of, say 500 horsepower a gas plant using fuels that make a tar-free gas will compare favorably with an oil engine or steam engine plant with its up-to-date auxiliaries. The larger installations need especial analyses in each case, and in very large installations the first cost is always higher than a steam turbine plant.

The cost of a small gas plant to use bituminous fuels will, with most of the types of apparatus available today, be entirely too expensive except in a very few cases, so that it is frequently best to put in a cheaper plant to handle anthracite coal.

When large bituminous plants are installed for the manufacture of clean gas for furnace work it is possible also to install gas-engines and make cheap power because of the fact that the power demand adds but little to the overall investment. A number of such installations have been in operation for years.

OPERATING COSTS

Fuel Consumption.—The fuel consumption of a gas plant may be figured on the following basis: Anthracite producers are from 65 to 80 per cent efficient but may be normally figured at an efficiency of 75 per cent.

Bituminous producers are from 65 to 75 per cent efficient depending upon the type.

The gas engine will consume from 9000 to 12,000 B.t.u. per b.h.p. hour. If, then, a given fuel contains 13,000 B.t.u. per pound as fired, the net B.t.u. delivered to the engine will be about 9375, and a b.h.p. delivered by the engine will require approximately 1.1 pounds of coal fired to the producer.

Labor.—Since the fuel consumption of a gas plant is less than that of a steam plant the labor required is proportionately less. The work necessary on an anthracite plant is that of charging fuel, which in some types of producers must be done either automatically or at frequent intervals, and with other types, due to the fuel storage within the equipment, only every three or four hours. The fuel bed must be barred down every six to ten hours and clinkers removed from the fire. Ashes are usually removed from the pit at least every twenty-four hours. It will be well to emphasize the fact that while the gas producer requires no more labor than the boiler plant the men in charge should be as competent as the engineer in charge of the steam plant of corresponding size. By giving all parts of the equipment skilled attention and supervision, looking after the saturation, cooling water, and particularly the gas-engine, better operating results will be obtained.

Water Required.—The amount of water required for cooling the gas depends upon the type of the producer and the fuel used

as well as the temperature of the water. The average plant will use from 3 to 4 gallons of water per pound of coal, although some plants will use as high as 10 gallons of water per pound of coal. This water may be recirculated over a cooling tower and the amount of water consumed thus reduced. When a recirculating system is installed for anthracite plants, increased water must be used to hold down the sulphur content of the water and prevent rapid sulphuric acid action throughout the recirculating system.

Standby Loss.—The standby loss on a gas producer is usually considerably less than that of a boiler plant and in no case should exceed 5 per cent of the total fuel fired when operating on intermittent service of ten hours per day.

Saturation—Steam or Vapor.—This item is not necessarily a charge against the operation cost, depending upon the source of supply. Some types of equipment have a saturator built in the apparatus and depend for the vapor on the heat that is ordinarily wasted through radiation. Others obtain heat by means of the exhaust gases of the engine, using them for the manufacture of steam or hot water. Still others use exhaust steam from a source of supply which is ordinarily wasted.

DESCRIPTION OF VARIOUS TYPES OF PLANTS

There are several types of anthracite producers on the market today which will be described in general, pointing out where possible the different methods used to obtain high efficiency and constant B.t.u. value and correct fire conditions.

Sharp-Bassett Producer.—This producer is shown in Fig. 257 and consists of a generator, down-comer, purge valve and scrubber or cooler. The generator is made up of a double shell, the inner shell being lined with fire brick and having step grates; although some of the late types have a grate with a hand-operated shaker. The space between the two shells is used as a saturator, the inner shell having several angle irons attached with holes at varying intervals for equal distribution of the air and contact with the hot water which is allowed to pass between the shells and thus absorb heat. There is an opening at the top of the shell and the air entering at this point picks up some vapor and is delivered to the under side of the grate. The method of obtaining increased or decreased saturation is by hand operation,

the producer depending for constant saturation temperatures under varying loads on the decreasing or increased amount of air entering between the two shells. This method is of necessity not a sensitive arrangement since fire conditions will not necessarily change as rapidly as the load conditions on the engine.

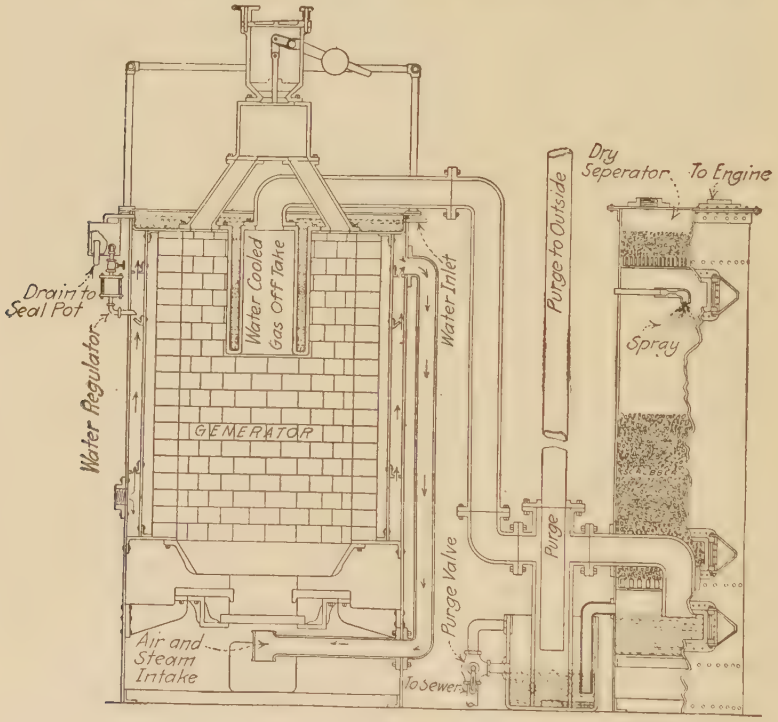


FIG. 257.—Sharp-Bassett Producer.

The scrubber is a cylindrical shell filled with coke and equipped with a water spray for cleaning and cooling the gas.

A modification of this type of plant is shown in Fig. 258. The equipment is in general as described above except that the space between the shells is designed as a boiler and some of the heat of the gas is also utilized for the manufacture of steam. This type of plant is more expensive than the straight producer and requires more attention. It is the writer's belief that with the modern highly efficient gas-fired boiler of today it is unwise to attempt to build an apparatus from which it is hoped to obtain

good results from both a gas-making and steam-making end of the equipment. It is difficult to find a place where the gas and steam demands may at all times be balanced so that the plant may easily be run under conditions that waste one or the other of the products with a resulting overall loss of efficiency.

Galusha Producer.—So far as the generator is concerned the Galusha producer is similar in design to a number of other makes

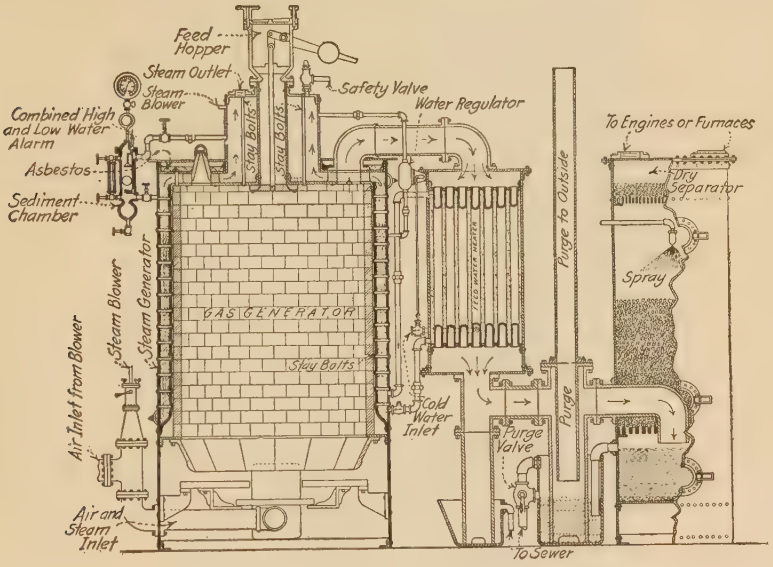


FIG. 258.—Sharp-Bassett Modified Producer.

of this type of apparatus, except for the fact that it is usually equipped with a brick-lined top and central offtake and surrounded by three charging hoppers. The central offtake, as will be seen from Fig. 259, is not equipped with central draft tube arranged for fuel storage within the producer itself. The grates of this producer are sometimes divided into sections, each section being provided with a hand lever for shaking.

The saturator consists of a steel pipe through which air and steam are admitted, although saturation for this type of producer is sometimes obtained by means of the exhaust from the engine as shown in Fig. 260.

The scrubber, or cooler, is a rectangular box arrangement equipped with a series of baffles and passage ways through which

the gas must pass and come in contact with water sprayed through the scrubber.

Smith Producer.—This producer is designed to utilize either anthracite coal, charcoal or coke. Several modifications are embodied in the producer equipment in accordance with the requirements found necessary for the fuels intended. The generator is built as shown in Fig. 261, with a central gas offtake,

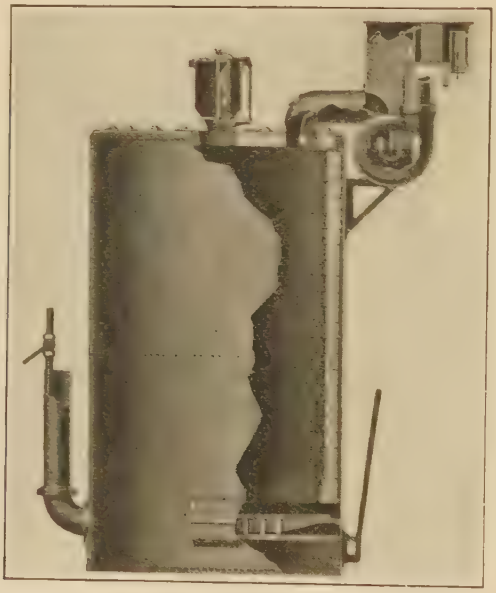


FIG. 259.—Galusha Producer.

the offtake being water cooled and with either three charging hoppers or a central hopper with distribution spouts. The space between the offtake, or draft tube, of the generator usually is used for fuel storage so that the producer may be run for a period of from four to six hours with one charging. The grates are always arranged for shaking either by hand or mechanical means. On the smaller units hand shaking is always used, while on the large equipment mechanical shaking is arranged for through either the use of compressed air or steam. The saturator shown in Fig. 262 permits the close association of steam and air in passing up through either large coke or cobble stones. This type of producer quite frequently receives its saturation, when used with

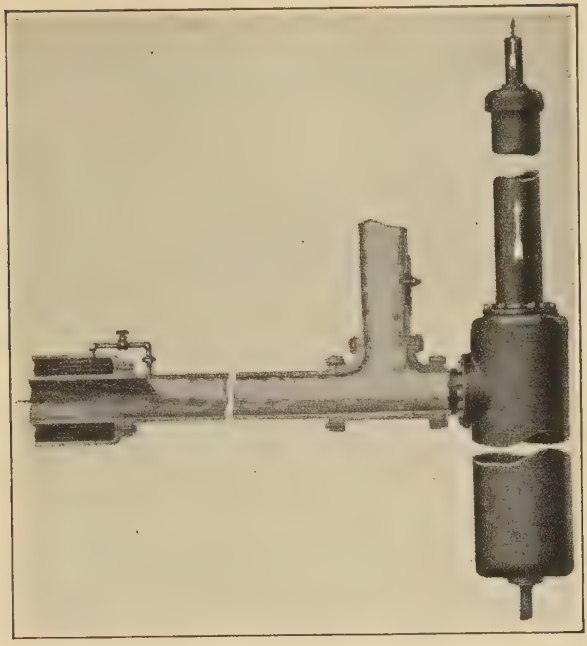


FIG. 260.—Saturator of Galusha Producer.

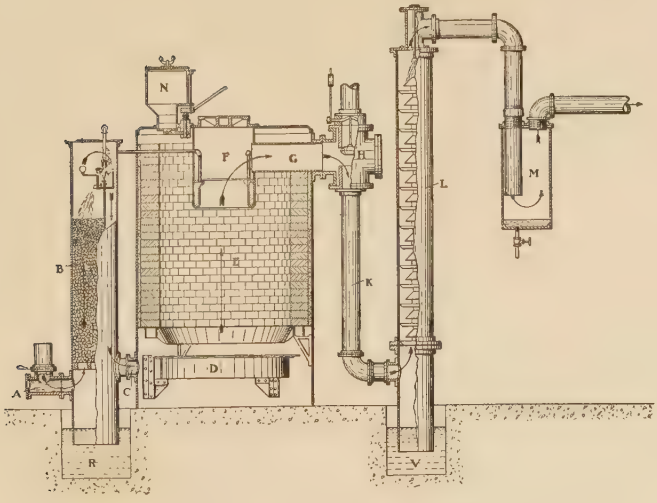


FIG. 261.—Section of Smith Producer.

a gas engine, by means of hot water obtained through what is known as the type-T heater shown in Fig. 263. This heater is attached to the exhaust of the gas engine. Exhaust gases passing up through the heater heats water passing through the tubes. From here the hot water is taken to the saturator and by means of a regulator of special design accurately controls the amount of hot water allowed to come into intimate contact with the ingoing air. This regulator may be set to obtain any degree of saturation desired and when so set will maintain such saturation throughout the varying loads. If steam is used for saturation the amount of steam required under varying load conditions to maintain a given saturation point is controlled by means of a steam regulating device in connection with the saturation temperatures.

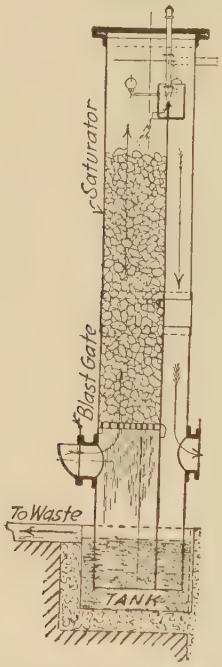


FIG. 262.—Section of Saturator, Smith Gas Engineering Co.

The condenser normally used when anthracite coal is the fuel is a baffle type condenser shown in Fig. 261. If it is found advisable to use either charcoal or coke, a second scrubber or purifier which consists of a cylindrical shell with removable trays and filled with sawdust and shavings is added to the equipment for a final scrubbing and cleaning of the gas. With charcoal there will be found some additional dust which will not always be removed by means of the ordinary cleaning apparatus, while with coke, regardless of the fact that it may often be lower in volatile by analyses, than anthracite coal, there is often a tarry vapor carried over with the gas which will not be removed by means of the ordinary baffle type condenser equipped with water sprays.

Smith Bituminous Producer.—The generator of the Smith bituminous producer is similar in design to that of the anthracite equipment shown in Fig. 264, except that it is normally equipped with a water-cooled head and a cylindrical charging hopper with distribution bell. The saturator is the same as that used on other types of Smith equipment and saturation may be obtained either by means of steam or hot water.

Because of the type of cleaning apparatus necessary the entire design and method of handling the gas after it leaves the generator are materially different than other types of machines gasifying either coke or anthracite coal. On leaving the generator the gas is carried to the bottom of a primary condenser through which the temperature of the gas is reduced to approximately

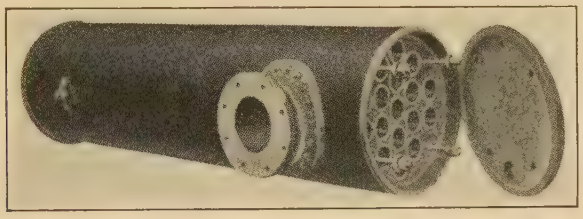


FIG. 263.—Smith Type-T Heater.

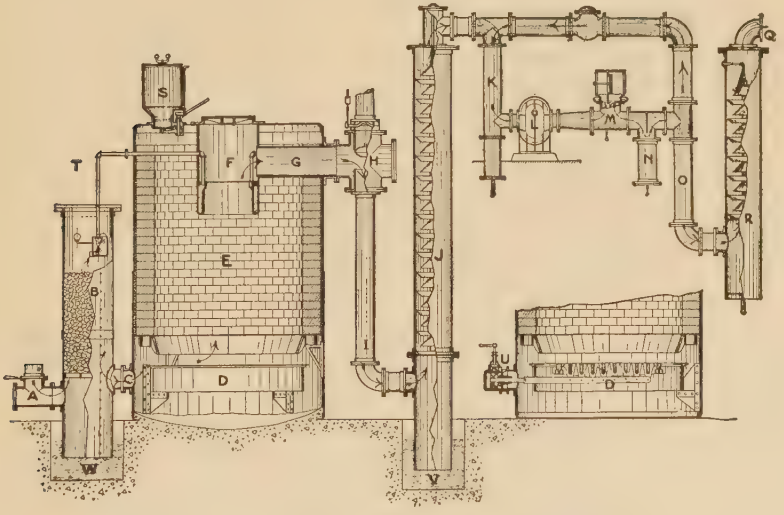


FIG. 264.—Diagram Showing Operation of Smith Type "BF" Plant.

125° F. on leaving the condenser. From here the gas passes to a positive pressure exhauster and, although some tar is brought down in the primary condenser, the lighter tar vapors which are more difficult to extract from the gas are carried through the exhauster and delivered to a special type of tar extractor at from $2\frac{1}{2}$ to 3 pounds pressure.

The type *F* tar extractor diagrammatically shown in Fig. 265 consists of a cast-iron box assembled over a gas main consisting of two compartments commonly known as the high- and low-pressure sides. Each box has assembled within it a glass wool mat and the gas delivered at high velocity through this mat is thoroughly cleaned of the remaining tar fog. Roughly, the principle upon which this mat operates is one of passing the gas laden with millions of tar particles through a glass wool mat and not

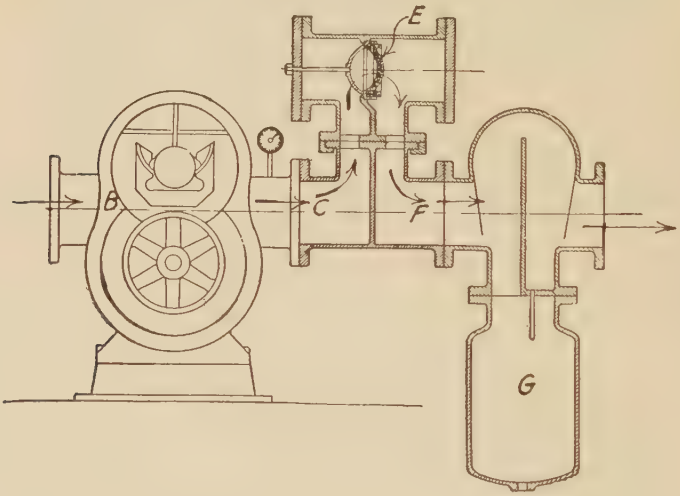


Fig. 265.—Diagram of Smith Type "F" Tar Extractor.

collecting the tar on the face of the mat, but through a colloidal action increasing the size of the tar drops sufficiently to allow them to drop out of the gas by specific gravity after leaving the mat. The tar mats may be of various sizes and several of them arranged on one header having ports in the header and the mat so that they may be cut in or out of service at the will of the operator. The power required for driving the exhauster is obtained from the gas engine itself, although the plant is equipped so that in starting and until after the first few revolutions of the gas engine these tar mats may be by-passed and the suction of the engine used for the delivery of the gas from the producer to the engine cylinders.

After leaving the tar mats the gas enters a secondary condenser where, by the means of baffles and water spray, the gas is cooled to as near atmospheric temperature as it is possible to cool it.

The tar which is extracted from the gas may, because of the method of extraction, be carried with the cooling water to a tar settling tank where the tar is allowed to settle from the water, and if advisable from an economical standpoint, the water may be recirculated over a cooling tower and again used for cooling within the condenser itself. The tar may easily be burned under the boiler or returned by patented methods to the producer itself and gasified. Such a scheme is being used on all the large bituminous producers of this type which are in operation today. The question of using this scheme for disposing of the tar will depend upon the amount of coal gasified since only 12 to 16 gallons of tar will normally be obtained from one ton of coal.

Gas Cleaning.—As stated earlier in this chapter cleaning of gas made from anthracite coal, at least, sufficient for gas-engine use is not a difficult problem, but gas made from bituminous coal contains millions and millions of tar particles many of them so small that they will pass through a felt hat, so that the cleaning of this gas presents an entirely different problem.

Many types of tar extractors have been invented, but they may be divided into four general classes: First, the bubbling washer in which the gas is brought into intimate contact with a washing fluid by being bubbled through it in finely divided streams. Tests for cleanness on this type of extractor run about as follows:

	Tar at inlet	Tar at outlet	Purification, per cent
Test 1	11.26	1.54	86.3
Test 2	12.92	1.44	86.8
Test 3	10.5	1.39	86.8

Second, baffle extractors in which gas passes through small openings and against opposed baffle surfaces. The following tests reported were on a tar extractor with six plates or baffles:

	Tar at inlet	Tar at outlet	Purification, per cent
Test 1	16.86	0.267	98.4
Test 2	15.15	0.302	98.0
Test 3	10.71	0.258	97.6
Test 4	15.21	0.133	99.1
Test 5	12.89	0.155	98.8
Test 6	15.56	0.420	97.3

Third, centrifugal tar extractors or washing fans. This type of extractor was used in this country a great deal in the early efforts to clean gas from bituminous coal, and one of the difficulties encountered was the unbalancing of the extractor. Tests are given below:

	Tar at inlet	Tar at outlet	Purification, per cent
Test 1	3.8	0.426	89.0
	13.8	0.458	96.7
	26.5	0.69	97.4

The above tests were made on equipment for cleaning gas for power purposes and give evidence that 98 per cent cleaning would be good practice. The writer's experience would indicate, however, that for such service a cleaning efficiency of not less than 99 per cent is necessary. A well-designed gas engine is not especially sensitive to tar and will function on gas that is not clean enough for long line distribution. Most gas engines are located near the producer plant. The velocity of travel of the gas is moderate and time elapsing between cleaning and using short, so that the light tar particles are often carried to the engine cylinder and burned. When distribution enters into the question greater care must be taken in cleaning the gas. The Smith type-*F* tar extractor, explained earlier in this chapter, will have a cleaning efficiency of at least 99.5 per cent.

Method of Testing for Clean Gas.—The method of testing ordinarily employed is to make weight determinations of a paper filter or extraction thimble before and after passing a measured quantity of gas through the filter. This method is quite satis-

factory for dust determinations where a large volume of gas can be passed and a large weight of dust collected, but it is not so satisfactory for tar determinations, since tar will soon block the filter and prevent large samples from being taken.

The process of drying to a constant weight before and after the sample is taken is a delicate operation and involves the use of chemical balances of high accuracy. These features serve to make this method a laboratory rather than a plant process.

The following method suggested by Mr. H. P. Smith has worked satisfactorily on plant tests for several years:

Filter paper holder of any convenient size, say 3 in. in diameter and an ordinary dry test meter reading down to .01 cu. ft., such as is commonly used by the City Gas Companies to determine gas consumption on various appliances is all that is necessary in the shape of equipment. With this apparatus connection is made to the discharge or clean gas side of a cleaning plant to be tested, and a sample of gas taken through a clean white filter paper. The filter holder should be placed with the paper horizontal and the flow of gas upward. The connection to the point of sampling should be as short and direct as possible. The rate of flow should be controlled by a valve placed on the discharge of the filter holder between it and the meter. If a valve is used in the connection between the point of sampling and the filter holder it must be wide open during the test. With a 3-in. filter paper gas can be passed at the rate of about 1 cu. ft. per minute. A higher rate may tear a hole in the paper. Ten cubic feet of gas should be passed through the first paper and this paper marked and laid aside for future reference. Then a second paper with 20 cu. ft., and a third with 30 cu. ft. should be taken in the same way. The connection should then be changed to the inlet or dirty gas side of the apparatus under test, and a series of filter papers taken starting with .01 cu. ft. for the first, .02 for the second and so on. Care must be taken on this series not to restrict the flow in any way until after the gas has passed through the filter. The connection between the point of sampling and holder must be short and the space filled with dirty gas before the paper is clamped into position. The quantity of gas passed through the second series of papers should be such as to make one of them match in color one of the clean gas series, and a little experience will show the correct volume for the conditions under test.

To obtain the final percentage of efficiency we will use the following example: Suppose that one of the dirty gas papers for .01 cu. ft. matches in shade the clean gas paper for 20 cu. ft.; this shows that there is as much tar in .01 cu. ft. of dirty gas as in 20 cu. ft. of clean gas, or as much dirt in 1 cu. ft. of dirty gas as in 2000 cu. ft. of clean gas. In other words, 1999 particles of tar have been removed out of a total of 2000 giving a cleaning efficiency of 1999 divided by 2000 equals 99.95 per cent. It will be noted that when the efficiency of the cleaning plant is high a large volume of gas must pass the filter of the clean gas series in order to produce the depth of color given by as little as .01 cu. ft. of dirty gas, so that for this reason the accuracy of the method does not change very much when very high cleaning efficiencies are being secured.

No definite color chart can be used for various quantities of gas because of the difference in color of the tars in different coals, so that in each case it will be necessary to take test papers on both sides of the tar extractor apparatus.

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